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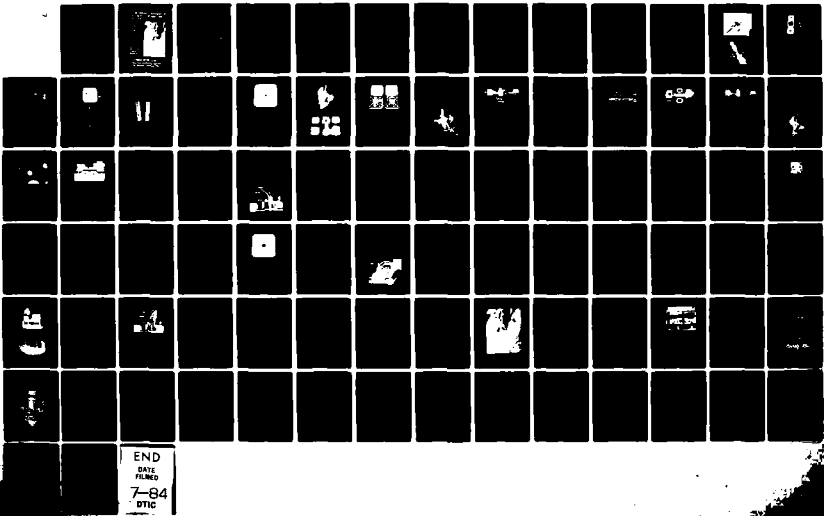

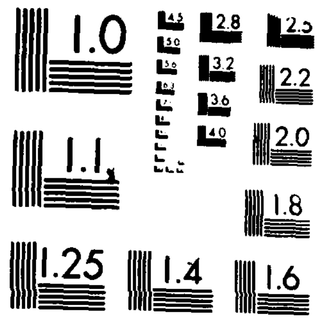

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Seawater Hydraulics: Development and Evaluation of an Experimental Diver Tool System

By

S. A. Black
K. Tate and
T. Conley

February 1984

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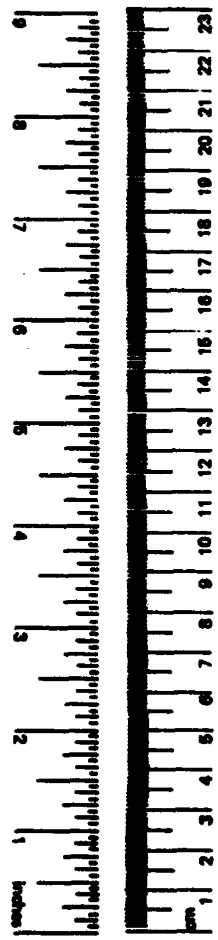
Naval Civil Engineering Laboratory
Port Hueneme, California 93043

METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2,000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
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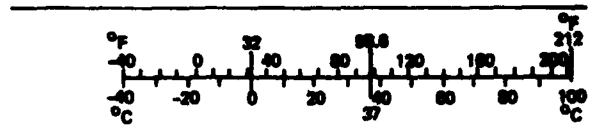
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1,000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



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The Naval Civil Engineering Laboratory has developed a diver-operated hydraulic tool system that uses seawater as the working fluid. The tool system consists of a diesel-driven hydraulic power source, a rotary impact wrench, and a rotary propeller cleaning brush. The rotary impact wrench is powered by an improved 3-hp reversible vane motor; the propeller cleaning brush is powered by a unidirectional vane motor. Both motors effectively provided

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The Naval Civil Engineering Laboratory has developed a diver-operated hydraulic tool system that uses seawater as the working fluid. The tool system consists of a diesel-driven hydraulic power source, a rotary impact wrench, and a rotary propeller cleaning brush. The rotary impact wrench is powered by an improved 3-hp reversible vane motor; the propeller cleaning brush is powered by a unidirectional vane motor. Both motors effectively provided power to perform the intended tasks. After 200 hours of operation, the reversible motor provided 3 hp of output at greater than 70% overall efficiency. Vane springs failed and were replaced after operating for 100 hours. The impact wrench delivered over 1,100 ft-lb of torque to a 1-inch-diam bolt in less than 7 seconds when used by a diver. Preliminary diver tests showed that the propeller cleaning brush functioned exceptionally well. The diesel-driven power source can deliver seawater at 12 gpm and 2,000 psi. This report presents the results of the test and evaluation of this experimental tool system.

CONTENTS

EXECUTIVE SUMMARY

This report describes the results of an RDT&E effort to develop an experimental seawater hydraulic tool system for use by Navy divers. The overall objective of this effort was to demonstrate the feasibility of using seawater rather than oil as the working fluid in diver tool systems. The Navy presently powers its hydraulic diver tools with oil. Seawater hydraulic systems are advantageous because they eliminate the need for oil; have a single hose umbilical; are compatible with the environment; reduce maintenance and increase reliability; and eliminate fire and health hazards.

An experimental seawater tool system has been successfully designed, fabricated, and evaluated. The basic design philosophy adopted for seawater tool systems was to provide at minimal cost a representative system suitable for laboratory and field evaluation by technical personnel. Several of the system's components were modified from commercial oil hydraulic components not specifically designed for seawater use.

The experimental seawater hydraulic system consists of a portable diesel hydraulic power supply, a rotary impact wrench, and a rotary propeller cleaning brush. A description of system components together with the test and evaluation results are contained in this report.

The seawater hydraulic power supply provides a portable, self-contained means of delivering pressurized seawater to the tools. The design was based on operational requirements for Navy construction divers. The power source is 192 inches long, 80 inches wide, 70 inches high, and weighs 3,500 pounds. The power source was tested and found capable of delivering flow rates to 12 gpm at pressures to 2,000 psi.

A rotary impact wrench is the most commonly used tool in the construction divers' tool kit. It provides the divers with a means of tightening and loosening threaded fasteners and drilling holes. The experimental impact wrench consists of: a reversible balanced vane motor; a pistol grip handle and valve mechanisms; and a rotary impact mechanism. The tool impact wrench is 13 inches long and weighs 17 pounds. Tests with the impact wrench proved that it is capable of delivering 1,100 ft-lb of torque to a 1-inch-diam-bolt. Divers had no difficulty in operating the impact wrench underwater.

Endurance tests were conducted with the reversible balanced vane motor that powers the impact wrench. The compact motor, which measures 3 by 3 by 3-1/2 inches and weighs 6-1/2 pounds, was operated for over 200 hours at 2.3 hp, 70% of rated power. At 100 hours, motor springs were replaced because of excessive wear. All other motor components showed little wear after 200 hours of operation.

The propeller cleaning brush was developed to demonstrate the use of seawater hydraulic tools for ship husbandry tasks. The brush is 11 inches in overall length and weighs 12 pounds in air. The brush is powered by a unidirectional balanced vane motor which consists of a pistol grip handle and valve mechanism and a nose piece for carrying radial and thrust loads generated from tool use. A preliminary evaluation of the brush proved satisfactory with only minor handling problems experienced by the divers.

Overall performance of the seawater hydraulic tool system was highly satisfactory. Results of the evaluation clearly demonstrate the feasibility of seawater hydraulic systems for powering diver tools. Operational and physical characteristics of both the reversible and unidirectional motors are well suited for diver tool applications. Further development is needed in the areas of: (1) a suitable seawater pistol grip handle with on-off and reversing valves; (2) improved vane spring life to provide greater than 200 hours of operation; (3) an improved bearing lubrication system for reversible motor operation; and (4) development of improved pressure-compensated controls for power source flow control.

INTRODUCTION

The Naval Civil Engineering Laboratory (NCEL) and the Naval Coastal Systems Center (NCSC) are engaged in programs to provide Navy divers with improved tools and techniques to enhance their capability to conduct work underwater. NCEL has the primary responsibility to develop equipment for construction divers who are engaged in the maintenance and repair of Navy seafloor and waterfront facilities. NCSC is responsible for equipment for Fleet divers who are engaged in salvage, repair, and maintenance of Navy ships. The research and development at both NCEL and NCSC has been conducted under the joint sponsorship of the Naval Sea Systems Command and the Naval Facilities Engineering Command.

A major part of the program to improve the working divers capability is directed toward the development of powered handtools. Most of the powered handtools presently used by divers, both commercially and within the Navy, are powered by oil hydraulic systems. While these tools provide the diver with the capability to perform many useful tasks, the need for oil to transmit the power from the source to the tool presents several disadvantages when used underwater. Many of the disadvantages result from the incompatibility of the oil-transmission fluid with the surrounding seawater. Leakage to and from the environment inevitably occurs, and the consequence can be either destruction of system components or pollution of the environment. Other disadvantages from using oil include the need for cumbersome, dual transmission hoses; fire hazards with the presence of oil in a high pressure oxygen environment; and potential health hazards for the diver when oil is present in the hyperbaric environment. A complete discussion of the advantages and disadvantages of oil hydraulic systems is contained in Reference 1.

An alternative to oil hydraulic systems is hydraulic systems that use seawater as the power transmission fluid. Research on seawater hydraulic systems has been underway at NCEL since 1976. A seawater hydraulic system would provide the diver with tools that are compatible with his environment and have all the attributes of the oil hydraulic systems now in use. Unique advantages arise from:

1. Elimination of the return hose and the use of a single smaller diameter hose system which is more flexible and easier for the diver to handle, and drag-imposed forces from ocean currents are significantly decreased.

2. Ability of the diver to connect and disconnect as well as assemble or disassemble the hydraulic components underwater because the working fluid is the environment.

3. Maintenance reduction because the tool functions with seawater internally as well as externally.

Early NCEL research in seawater hydraulic systems resulted in the successful development on an experimental, unidirectional balanced vane motor (Ref 2). This motor produced 3.2 hp at 1,600 rpm with a seawater input flow of 7 gpm at 1,000 psi. The experimental motor operated for 50 hours at full power before the vane springs failed.

Success with the experimental vane motor proved the feasibility of using seawater as the working/lubricating fluid for small compact motors and paved the way for future development.

With the experimental mctor as a starting point, an experimental seawater hydraulic tool system was developed on contract. The overall objective of this work was to produce a complete system that could be used as a test bed to demonstrate the feasibility of developing seawater hydraulic systems for diver use. The three major tasks of the contract development effort included designing and fabricating:

1. An improved version of the balanced vane motor to include reversible operation and longer service life.

2. A hand-held, diver-operated rotary impact wrench powered by the improved vane motor.

3. A portable, diesel-driven seawater hydraulic power source.

The experimental seawater hydraulic tool system was delivered to NCEL in January 1982. This report provides a description of the tool system and the test and evaluation results. A complete description of the system development is contained in Reference 3.

In January 1982, NCEL and NCSC initiated a joint project to develop and evaluate a propeller cleaning brush that was powered by the seawater hydraulic vane motor. The overall objective of this project was to demonstrate the practicality and feasibility of seawater hydraulics for use in ship husbandry tasks and to encourage wider acceptance of the seawater-powered diver tool concept. NCEL was responsible for the design and fabrication of the tool, and NCSC was responsible for test and evaluation. The results of the design, fabrication, and initial testing of the propeller cleaning tool are contained in this report.

SYSTEM DESCRIPTION

The design philosophy adopted for developing the seawater hydraulic tool system and the propeller cleaning brush was that of providing at minimal cost a representative system suitable for laboratory and field evaluation by technical personnel but without the ruggedness and life cycle requirements needed by Fleet divers. This system would then be evaluated to demonstrate the feasibility of seawater hydraulic systems and identify developmental problems that could be solved prior to developing a prototype system. Several of the components of the impact wrench, power source, and propeller cleaning brush were adapted from commercial oil hydraulic systems and were not specifically designed or optimized for long-term diver or seawater use. The next stage in the developmental cycle will be to develop a prototype system that is compatible with the diver and seawater.

The portable power source and the impact wrench were developed simultaneously under contract (Ref 3). After these components were developed and while they were being evaluated at NCEL, the development of the propeller cleaning brush began. Thus, some of the problems encountered with the impact wrench were solved and incorporated as design changes in the propeller cleaning brush.

Because the impact wrench is used for both tightening and removing nuts, it is required to operate in both forward and reverse directions. The propeller cleaning brush operates in one direction only. Two motors were developed to provide these capabilities. The basic difference between the reversible and the unidirectional motor is the method of lubricating the bearings. As described later, unidirectional operation greatly simplifies bearing lubrication design and results in a lighter, more compact motor.

Improved Reversible Vane Motor

Description. A summary of the performance and operational design goals selected for the improved motor is given in the following table.

Performance Item	Goal
Power, hp	3
Efficiency, %	75
Minimum operating pressure, psi	1,200
Maximum water weight, lb	7
Service life, hr	250
Maximum flow, gpm	7
Maximum volume, in.^{3}	25

These were based on a comprehensive analysis (Ref 1) of operations and environmental constraints imposed on the diver. Since the heart of the diver tool system is the motor, it is important that it be compact, lightweight, and efficient.

A major portion of the development effort was to redesign the experimental motor for reversible operation, longer service life, and compatibility with the components required for a diver-operated rotary impact wrench. The improved motor is shown assembled in Figure 1 and disassembled in Figures 2 and 3. The basic design is that of a balanced vane type in which pressure is applied simultaneously to radially opposite ports. In this way, no hydrostatic loads are applied to the bearings, and their design is greatly simplified.

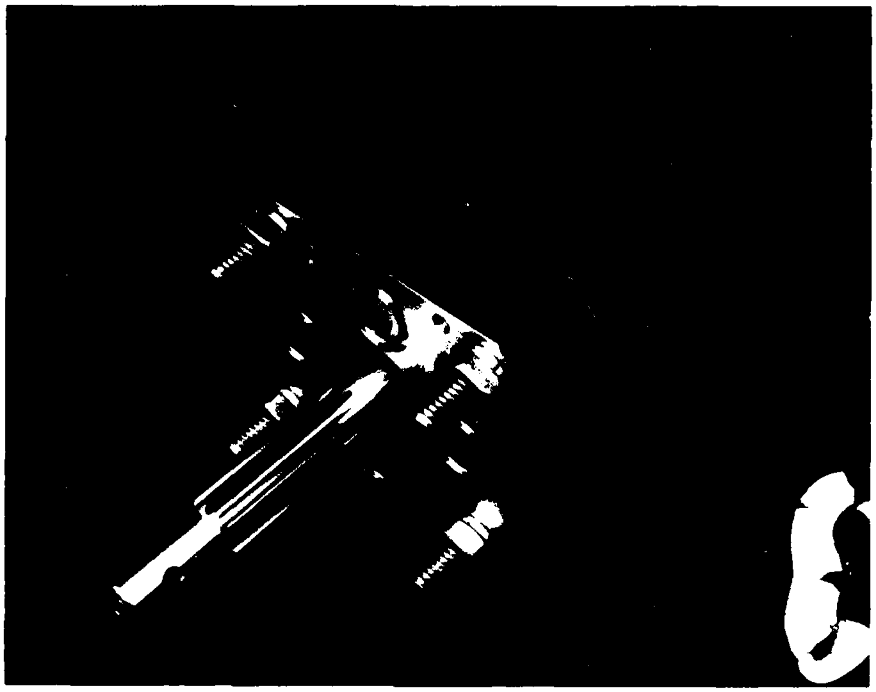


Figure 1. Assembled seawater hydraulic vane motor showing dual inlet and outlet ports which mate with impactor ports. The motor measures 3x3x3-1/2 inches long and weighs 6-1/2 pounds.

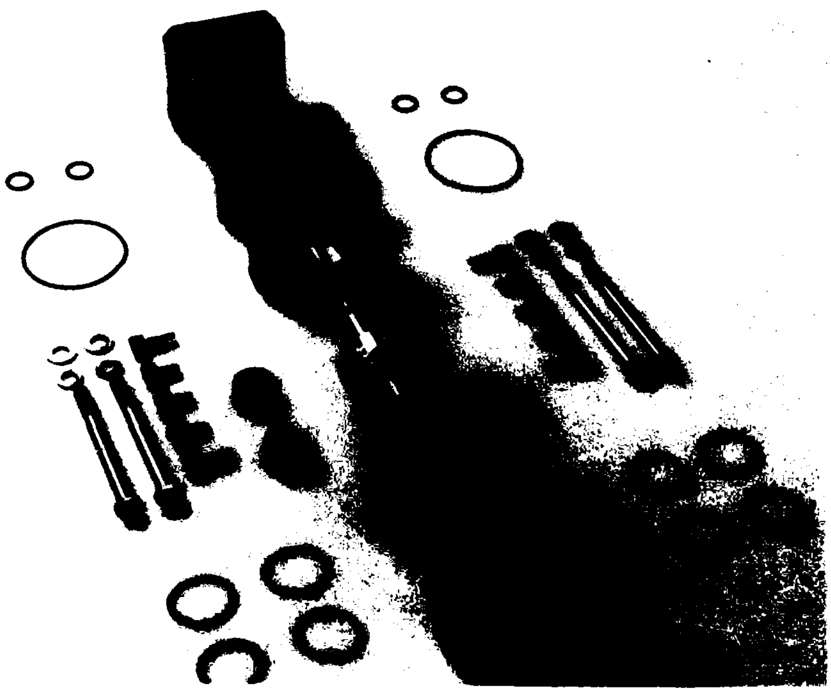


Figure 2. Disassembled vane motor showing all components including 10 vanes and 20 springs.

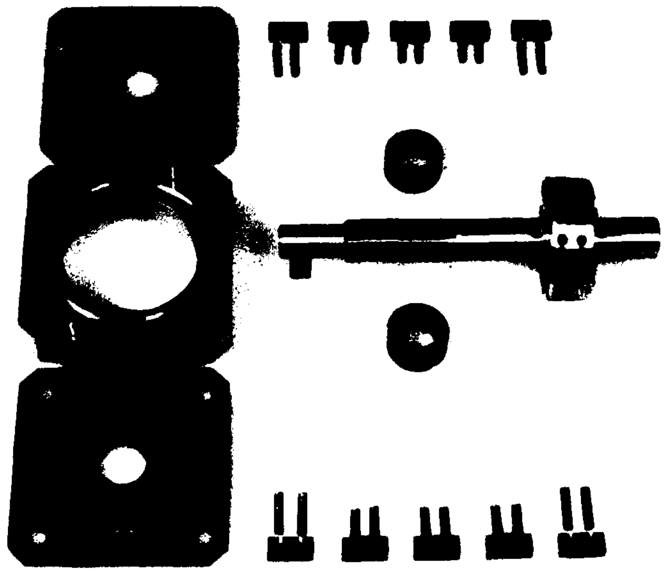


Figure 3. Disassembled vane motor showing high wear components: vane, rotor, track, flexible side wear plates, and vane springs.

The operational components of the motor include 10 vanes; two end plates; two flexible side plates; vane track; rotor shaft; and 20 springs. The rotor shaft, end plates, and cam are fabricated from Inconel 625*. Bearings, vanes, and side plates are fabricated from Torlon 4275, a high strength, thermo-plastic material with graphite and Teflon fillers. Vane springs are fabricated from Elgiloy wire, a cobalt-nickel-based alloy.

The shaft and rotor are made in one piece. Splines on the end of the output shaft are designed to mate with an adapter shaft which drives the impact mechanism. The rotor has slots for 10 vanes with two spring-retaining holes for each slot. A hole is drilled axially through the shaft and plugged at both ends. Radial holes are drilled completely through the shaft on each side of the rotor (Figure 4) to meet the center hole. As described later, these holes provide a path for seawater flow to lubricate and cool the bearings.

Two plastic side plates, located on either side of the rotor, contain four sets of ports for fluid control. The outer set of ports, which sweep through 40 degrees, direct fluid to and from the working chambers to provide rotary motion and torque. The inner set of ports sweeps through 30.6 degrees and serves two functions: first, they prevent the trapping of fluid below the vanes; and second, they provide a means of applying pressure below the vanes to help counteract the radially inward forces generated at the vane tips.

*Information on the mechanical and physical properties of some materials discussed in this report are contained in the Appendix.

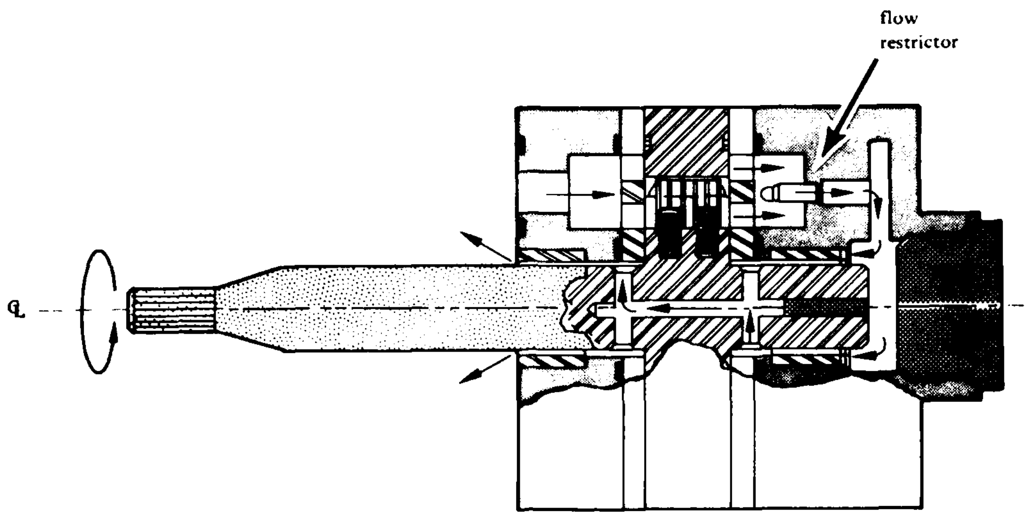


Figure 4. Bearing lubrication path. Flow restrictor allows high pressurized seawater to lubricate bearings.

The side plates are designed to be flexible to help provide a seal at the side of the rotor and at the edges of the vanes. The side plates also serve as thrust bearings for the rotor. Flexing, or movement of the side plates against the rotor, results when seawater pressure is applied to the inlet ports. The pressure area is defined and bounded by the O-ring seals shown on the end plates in Figure 5. The thickness of the side plates was selected to provide the correct amount of flexing at the motor operational pressures.

The front and back end plates provide support for the shaft sleeve bearings. The front end plate also contains the fluid inlet ports while the back ports contain flow restrictors that provide the seawater for bearing lubrication and cooling. One flow restrictor is installed in each set of ports (Figure 5).

The Inconel cam track defines the radial motion of the vanes and provides the sealing and wear surface for the vane tips. The ramp shape is an approximation of a cycloidal shape tangent to the minor (0.95-inch) and major (1.05-inch) radii. The cam track profile was selected to minimize accelerations of the vanes. Each vane cycles twice per revolution and has a radial movement of 0.1 inch. O-rings on either side of the vane track provide sealing between the track and the flexible side plate.

To minimize wear at the tip of the vane, the tip has a symmetrical radius as large as the maximum slope of the vane track ramps would allow. Two spring holes are provided in each vane. The groove on each side of the vane is designed to provide a high-pressure leakage path to allow balancing of the pressures between the top and the bottom of the vane. Figure 6 is a vane designed with the high-pressure leakage path.

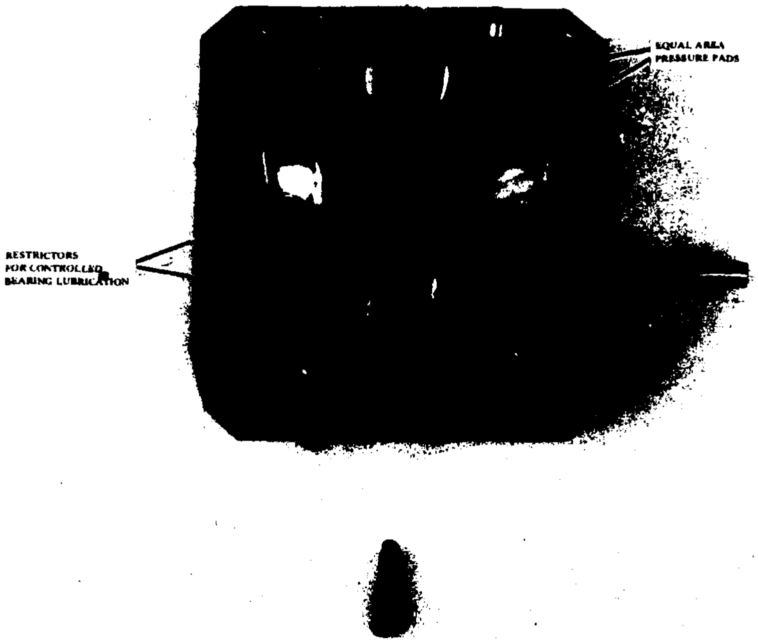


Figure 5. Aft housing pressure pad areas showing flow restrictors.

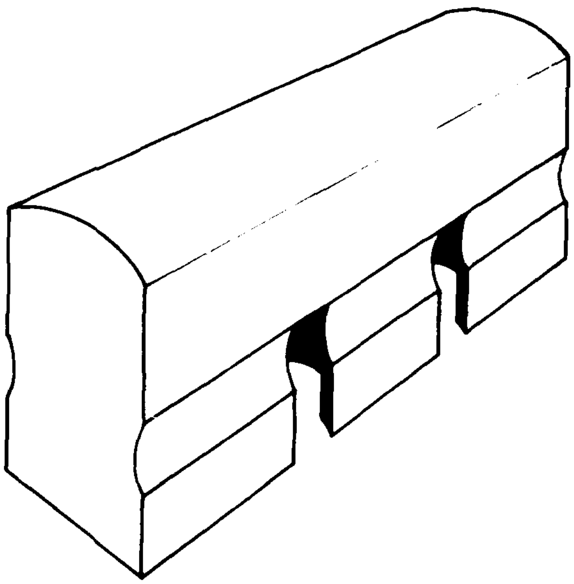


Figure 6. Vane with high-pressure leakage path (path helps to equalize pressure at the base of vane in the rotor).

Motor Improvements. Increased service life and reversibility were of particular importance in the improved motor design. The experimental motor described in Reference 2 ran for 50 hours at full power output before the vane springs failed, resulting in reduced motor performance. It was thought that the spring failure was due to wire fatigue; however, it was later learned that, although fatigue failure was a problem, the springs were also rubbing against the sides of the holes in the Inconel rotor. The metal-to-metal rubbing caused heavy wear and loss of material from several coils in each spring (Figure 7).

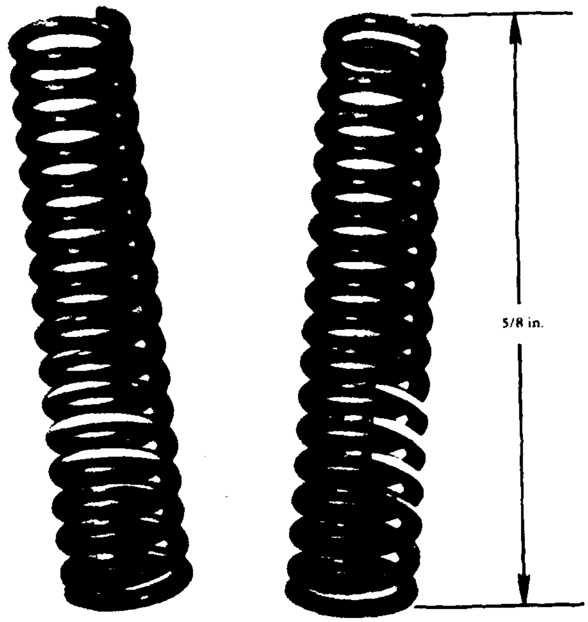
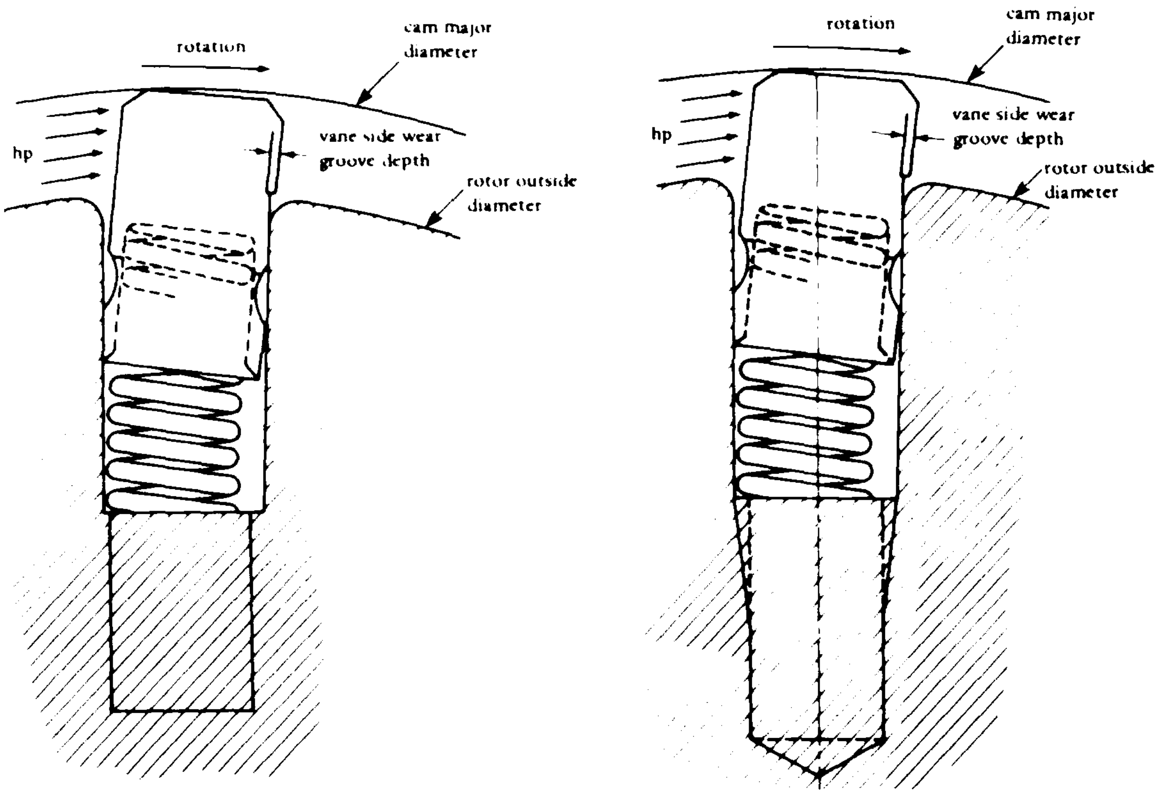


Figure 7. Heavy wear and loss of material on a vane spring due to rubbing contact with rotor.

In order to improve the spring's working life, an investigation was conducted to determine the optimum spring force, geometry, and material for construction. The optimum force was determined experimentally to be 0.75 pounds for the motor tested. Later the design was changed to to 1.25 pound to accommodate increased vane mass. To further increase fatigue resistance, the wire and spring outside diameters were increased and the spring material changed from 17-4 PH stainless steel to Elgiloy wire. The final spring design included a shorter free length to reduce column length and minimize column buckling.

The magnitude of the spring wear problem was not realized until spring fatigue resistance was increased beyond 50 hours. The overall problem involves an interrelationship between the rocking motion of the vane in the vane slot, vane side wear, spring motion, and spring buckling. A sketch of the cross section of a vane slot and spring hole for the experimental and improved motor is shown in Figure 8. Initially, a clearance of 0.003 inch exists between the vane and vane slot. The vane cocks slightly when pressure is applied. The contact area on the leading face of the vane with the rotor is relatively small and high PV loads result in subsequent vane-face wear. The wear results in additional cocking of the vane to a point where spring motion and buckling causes the spring to contact the rotor. To help relieve this problem, the spring hole was tapered as shown in Figure 8.



(a) Old vane slot and spring hole

(b) Tapered spring hole

[illegible]. Comparison of spring holes. The tapered hole (b) tended to wear substantially less on the spring.

In g[illegible]ersing the rotational direction of a vane motor simply invol[illegible]ing the pressurized seawater supply to the outlet ports. Fluid [illegible] applied to the opposite side of the vane and the rotation direction is changed. In the experimental motor, bearing lubrication was provided by flow channels in the flexible side plates from the low pressure ports as shown in Figure 9. The slots in the side plates mate with axial slots in the bearings. Applying high pressurized seawater to the reverse ports results in a high volume leakage path.

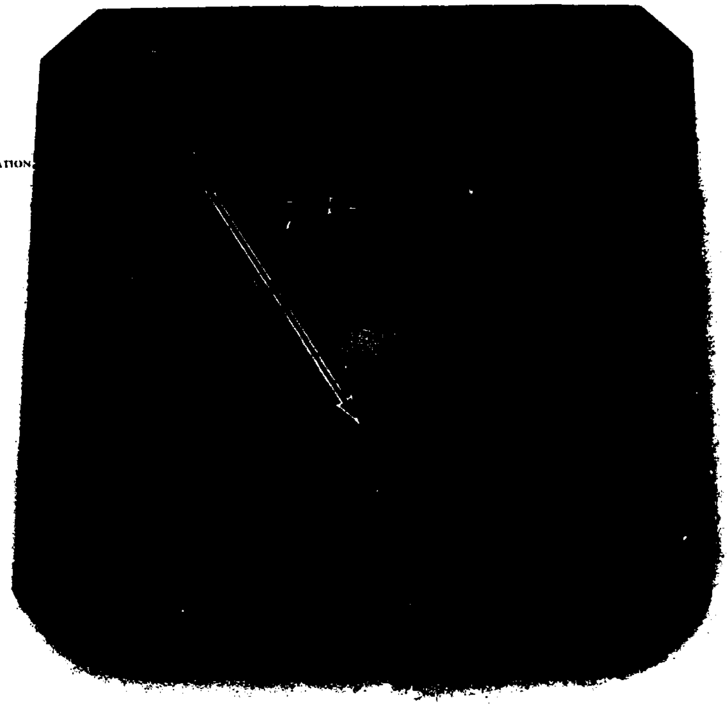


Figure 9. Experimental motor Torlon side plate with lubrication slot shown.

To provide bearing lubrication for the reversible motor, a flow restrictor was installed in each set of ports in the motor's back plates as shown in Figure 5. Pressurized seawater applied to the restrictor allows seawater to flow, as shown in Figure 4, through axial slots in the rear bearings, through the rotor shaft passage, and then out of the motor via the axial slots in the forward bearing.

Unidirectional Motor

A modified version of the improved vane motor was used to power the propeller cleaning brush. First, the motor was converted from bi-directional operation to unidirectional operation. This was done to simplify the water ducting system that lubricates the motor bearings. Second, the seawater return porting in the motor was eliminated, and the water was exhausted directly to the environment. This was done to simplify porting and valving in the tool handle and to minimize flow and pressure losses within the handle.

The unidirectional motor is shown assembled in Figure 10 and disassembled in Figure 11. The motor measures 3 by 3 by 3 inches and weighs 4.5 pounds. Most of the functional components of the unidirectional motor are similar to the reversible motor. Front and back end plates for the motor are fabricated from 316 stainless steel. The flow restrictors were removed from the motor's back plate and the thickness of the back plate reduced to minimize motor weight. Two pie-shaped sections were removed from the front end plate at the exhaust ports where it mates with the flexible side plate (Figure 12). This was done to accommodate direct exhausting of the low pressure seawater at the motor.

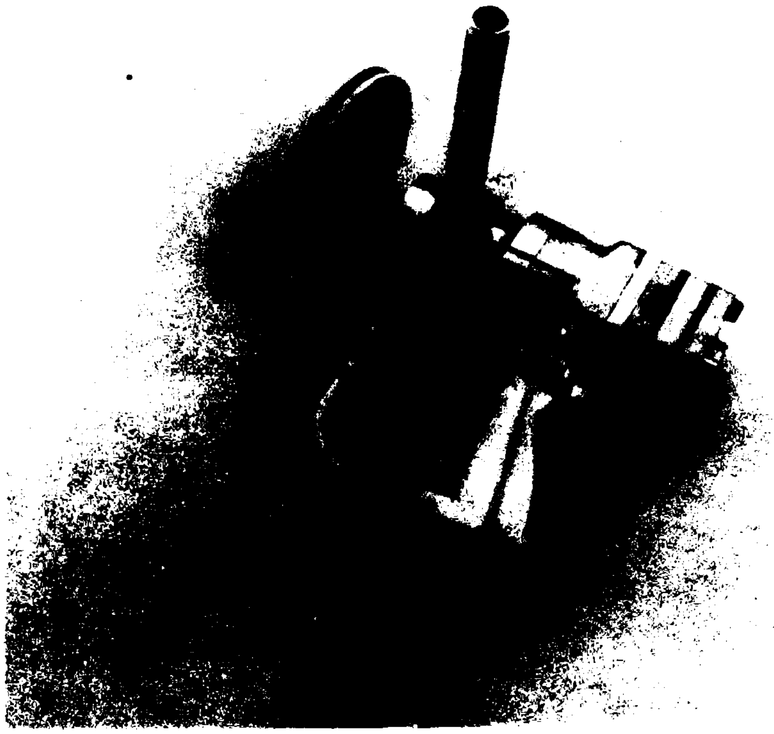


Figure 10. Assembled propeller cleaning brush with unidirectional motor.

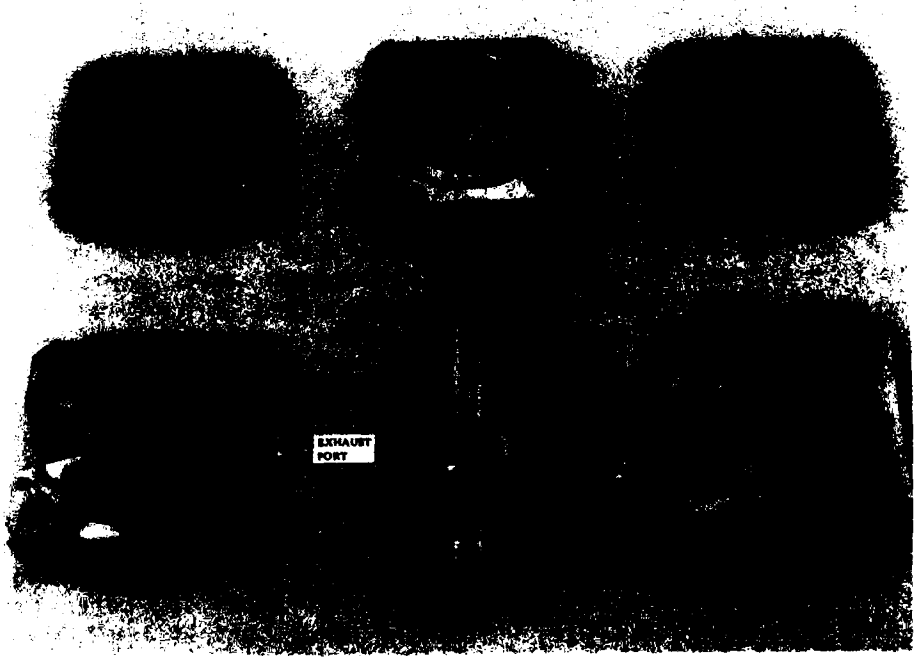


Figure 11. Major components of unidirectional motor. Note cut out for the exhaust port.

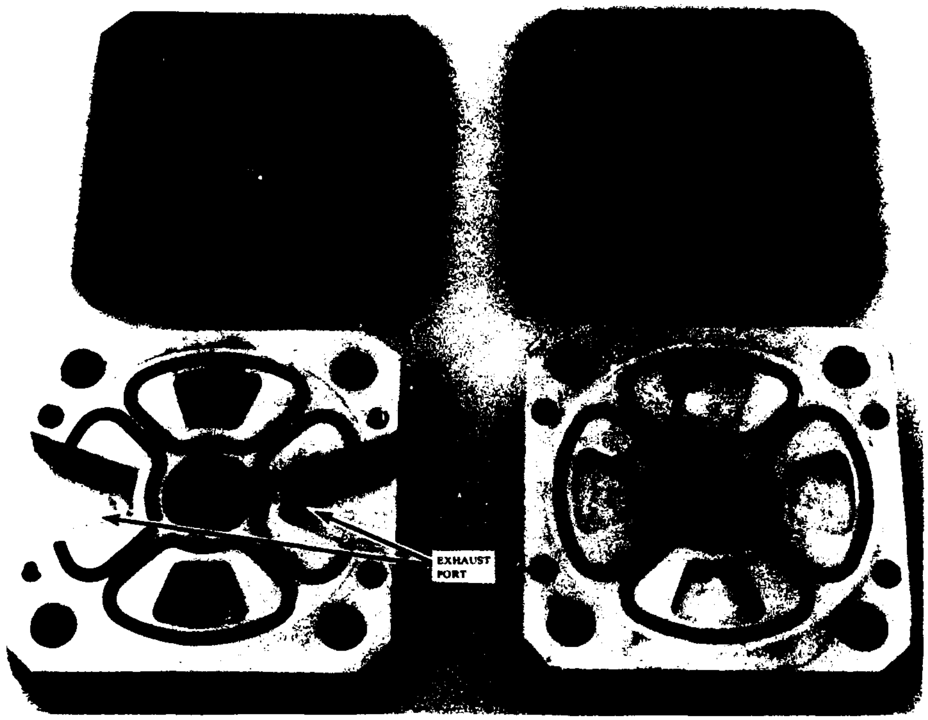


Figure 12. End plates and side wear plates for unidirectional motor with exhaust port cut outs.

The central bearing lubrication channel was removed from the rotor. Bearing lubrication was instead provided by channeling fluid from the low pressure exhaust ports directly to the bearings. This was accomplished by flow channels in the flexible side plates as shown in Figure 9.

The spring holes in the rotor were machined with straight sides rather than tapered as in the reversible motor. This was done as an experiment to determine the effect on spring wear and to simplify machining to reduce fabrication cost. The rotor output shaft was equipped with a tang-type end to mate with a slot on the propeller brush adapter shaft.

A porting adapter plate was fabricated from aluminum to direct fluid from the handle pressure port to the motor inlet ports. Since the motor is unidirectional and fluid is exhausted directly at the motor, the adapter plate design was greatly simplified over that of the reversible motor.

Rotary Impact Wrench

General Description. A rotary impact wrench was chosen as the first tool to be developed because it is the most widely used tool in the construction diver's tool kit. The impact wrench is used for tightening and loosening threaded fasteners as well as for drilling holes. Several of the impact wrench components were modified parts from terrestrial oil hydraulic tools in order to keep costs at a minimum. Later development of the wrench will require that these components to be designed to be fully compatible with the diver, the seawater environment, and the seawater working fluid.

The complete experimental impact wrench shown assembled in Figure 13 and disassembled in Figure 14 weighs approximately 17 pounds and is 13 inches long. The major components of the wrench include the motor, pistol-grip handle and valve assembly, handle adapter plate, and impact mechanism.

A schematic of the seawater flow in the impact wrench is shown in Figure 15. Fluid is directed via the on-off and reversing spool valves to one set of dual ports on the motor. Discharge flow from the motor exits the handle adjacent to the inlet port. Seawater from the forward motor bearing is directed axially along the shaft to cool and lubricate the radial and thrust bearing located in the handle adapter plate. This bearing serves to prevent thrust and axial loads from being transmitted to the motor. Bearing flow exits the tool at the forward impactor bearing.

Impact Mechanism. A primary concern during the development of the impact wrench was the selection of a suitable impact mechanism. Two different mechanisms were considered, both manufactured by Ingersoll-Rand (IR). The IR Model 5100 is used on conventional oil hydraulic tools presently being used by Navy divers. The IR Model 2910 is coming into greater use because of higher efficiency, simpler design, and increased reliability.

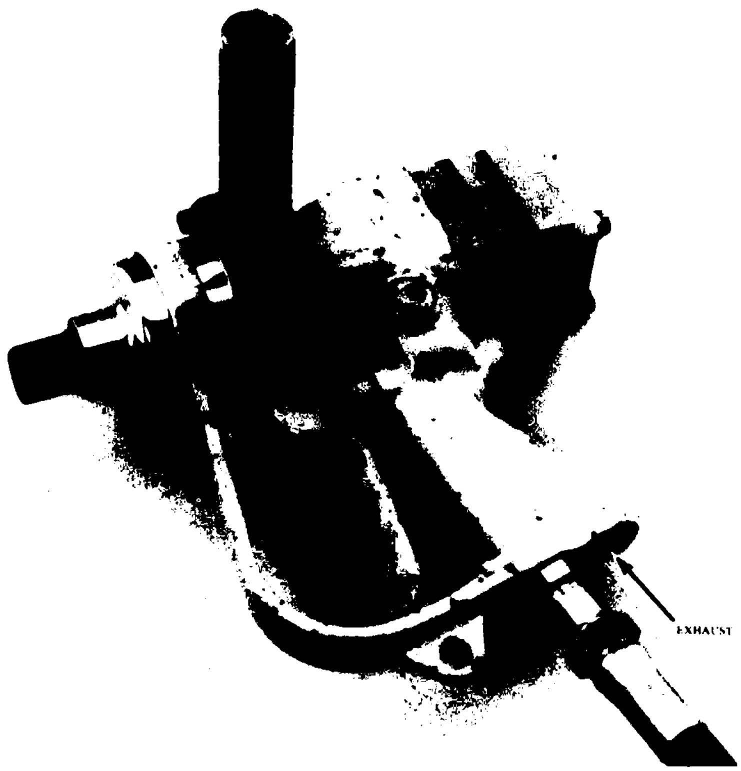


Figure 13. Assembled experimental impact wrench. Note the single inlet hose. Fluid was exhausted along side the inlet hose as shown.

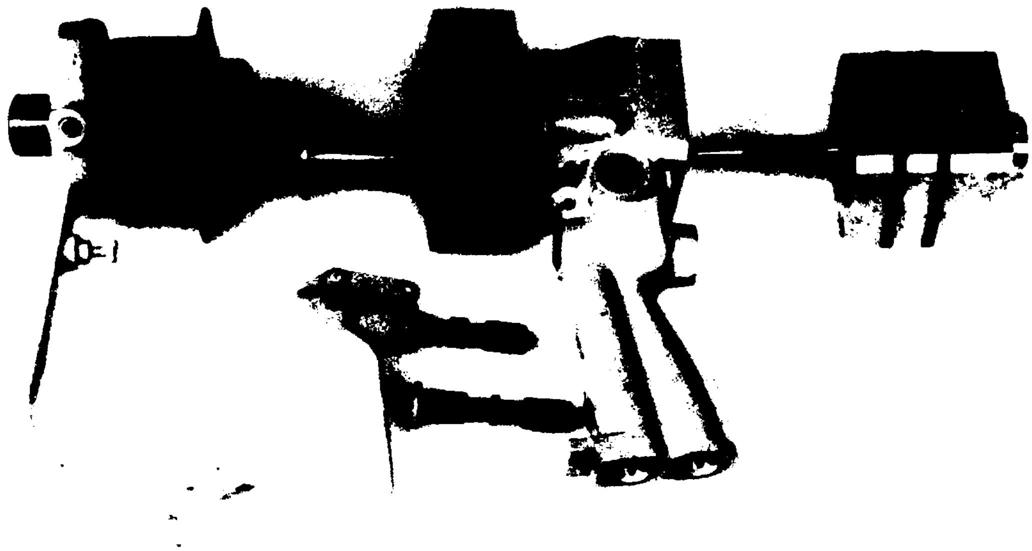


Figure 14. Impact wrench showing major subcomponents.

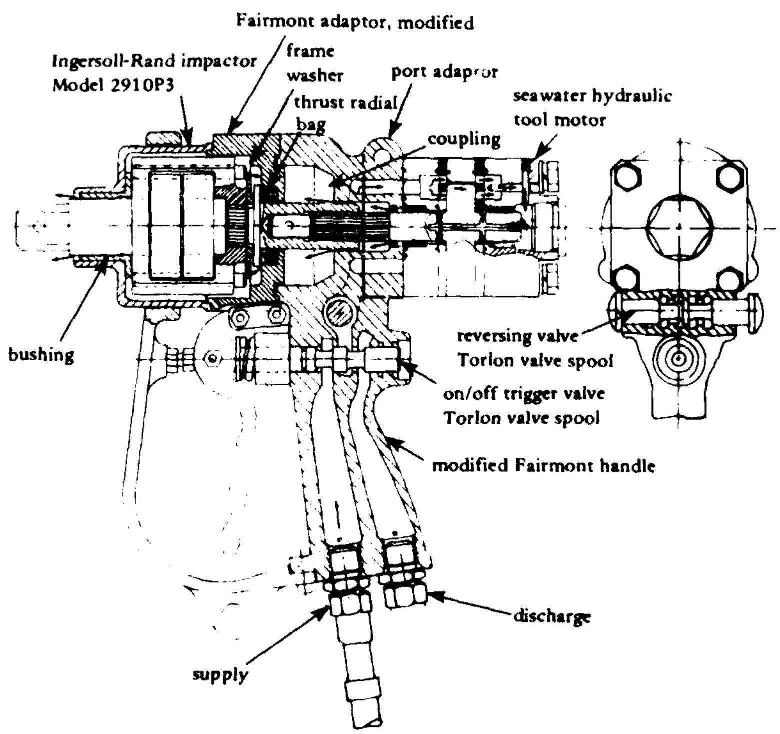


Figure 15. Impact wrench showing seawater fluid path for bearing lubrication within the impact mechanism.

The IR Model 5100 (Figure 16) is coupled to the motor through a planetary gearbox and coiled compression hammer spring. The spring mates with the cylindrical hammer; motor rotation causes the spring, hammer and anvil to rotate. Resistance applied to the anvil-output shaft is transmitted through the anvil dogs to the hammer dogs. At a predetermined output shaft resistance, the hammer moves axially rearward until the dogs disengage. Following disengagement the spring potential energy causes the hammer to accelerate (both radially and axially) via the cam balls until the hammer and anvil dogs engage. At this moment, the hammer's kinetic energy is transmitted to the anvil. This cycle is repeated twice for each revolution of the gearbox output shaft. The planetary gear arrangement provides a 9:1 reduction from the motor output shaft speed.

A major problem with the Model 5100 impact mechanism is that it is subject to damage from excessive rpm's (overrevving), a condition that occurs frequently with diver use. When this occurs, chips of metal occasionally break away from the hammer and anvil dogs and become lodged between the planetary gear teeth. The effect is either a jammed mechanism or broken teeth. While it may seem prudent to limit system flow rates to prevent overrevving, this is not always done.

The Model 2910 impact mechanism was selected for use because of the frequent failures experienced with the Model 5100 impact mechanism. In addition, the design of the Model 2910 impact mechanism is more rugged and has fewer moving parts. The Model 2910 (Figures 17 and 18) uses two hammers that are connected indirectly via drive pins attached to the hammer frame. The hammer frame is rotated by the motor output shaft. Centripetal forces hold the hammers in an off-centered position. When a load is applied to the output shaft the hammers simultaneously strike the anvil once at each revolution. At impact, the hammers' kinetic energy is transferred to the output shaft anvil and the hammers cam radially off the anvil. This motion is such that the motor stops once each revolution at impact.

Minor modifications were made to the off-the-shelf impact mechanism to improve its underwater performance. All steel parts were coated with Lubriplate* to minimize corrosion. The parts were immersed in boiling Lubriplate at 400°F to increase penetration.

The start-stop characteristics of the Model 2910 impact mechanism were particularly worrisome during the early impact wrench design stages, since it was not known whether the seawater motor could accelerate rapidly enough to be effective. As a possible solution to this problem, a flexible coupling was designed for mounting between the motor and the impactor. When the impact mechanism stopped rotating, the motor would continue to rotate and deform the flexible coupling. Later testing of the motor coupled to the impact mechanism showed that the flexible coupling was not necessary.

*Lubriplate is an extreme pressure-type lubricant composed of zinc oxide grease and lithium soap.

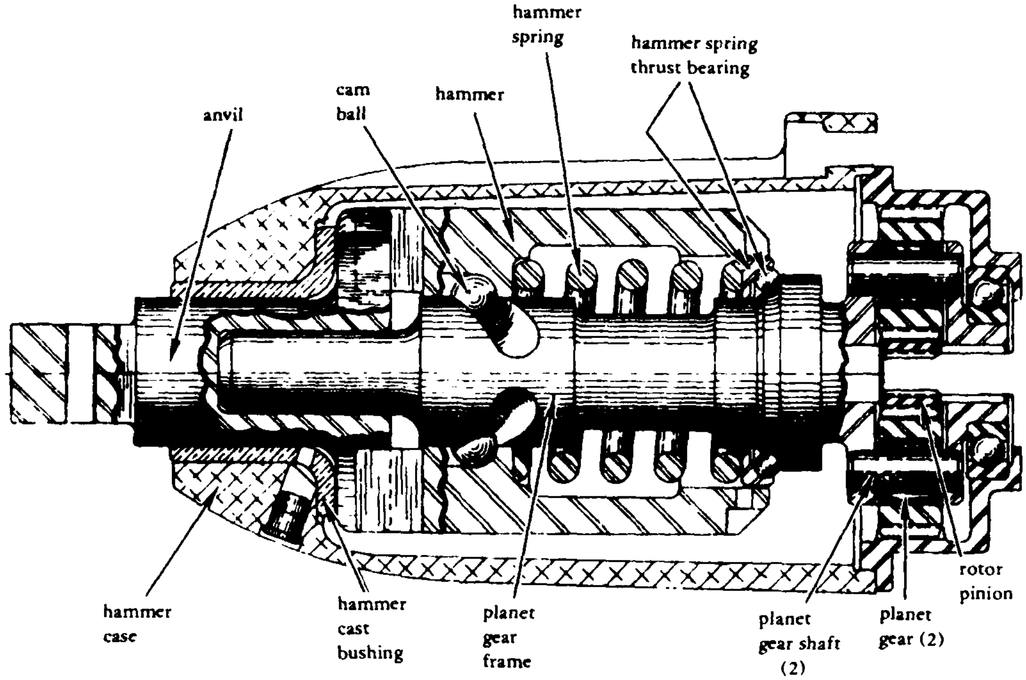


Figure 16. Ingersoll-Rand's Model 5100 impact mechanism as configured on presently used oil-hydraulic tools.

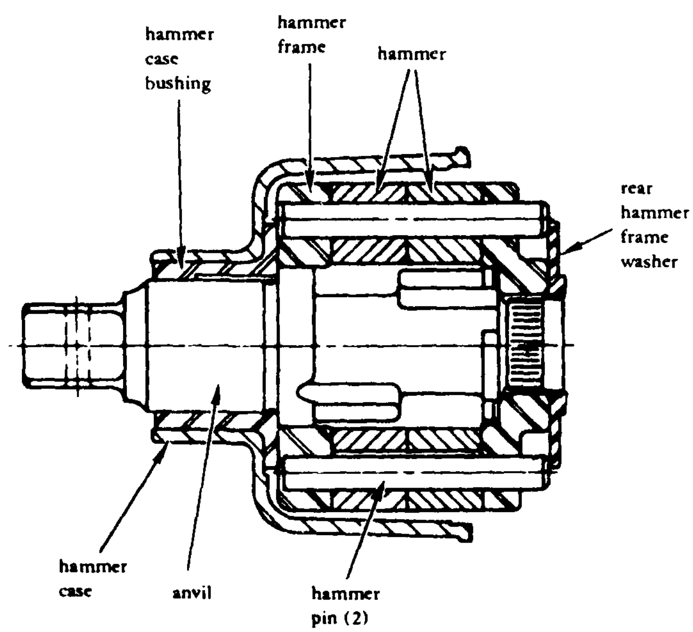


Figure 17. Arrangement of components of Ingersoll-Rand's Model 2910 impact mechanism.

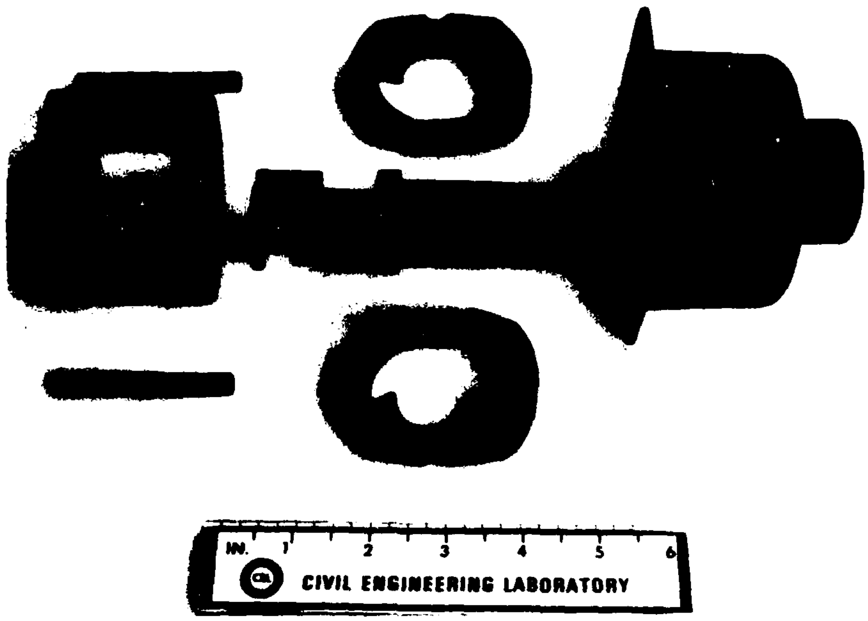


Figure 18. Major components of IR Model 2910 impact mechanism.

Rotary Impact Wrench Handle Assembly. The handle assembly of the wrench (Figure 19) consists of the handle body, two spool valves, trigger mechanism, trigger guard, and adapter plate for attaching the impact mechanism to the handle. Since the experimental wrench was intended only for laboratory and field use by technical personnel, it was fabricated from a modified commercially available oil hydraulic tool and was not used in seawater for long periods of time. The cast aluminum handle was manufactured and modified by Fairmont Railway Motors, Fairmont, Minn., from one of their standard tool models.

The Fairmont handle was modified by changing the shape at the back of the handle to accommodate the four mounting bolts of the vane motor; fabricating a port adapter plate to direct fluid to the motor ports; and installing stainless steel self-locking threaded inserts for mechanical attachment of the motor. The port adapter plate was cast from aluminum in two pieces and then fastened together and to the handle with threaded fasteners and a high strength, resin-based adhesive to provide sealing at the butt-joint.

The two spool valves in the handle provide on-off and forward-reverse operation. The sleeves for the spools were precision-bored directly in the aluminum handle casting. The spools were initially fabricated from Torlon plastic, but dimensional changes from swelling of the plastic caused the spools to be inoperable. Subsequent spools were fabricated from stainless steel and flame-sprayed with a molybdenum coating to minimize corrosion of the aluminum bore.

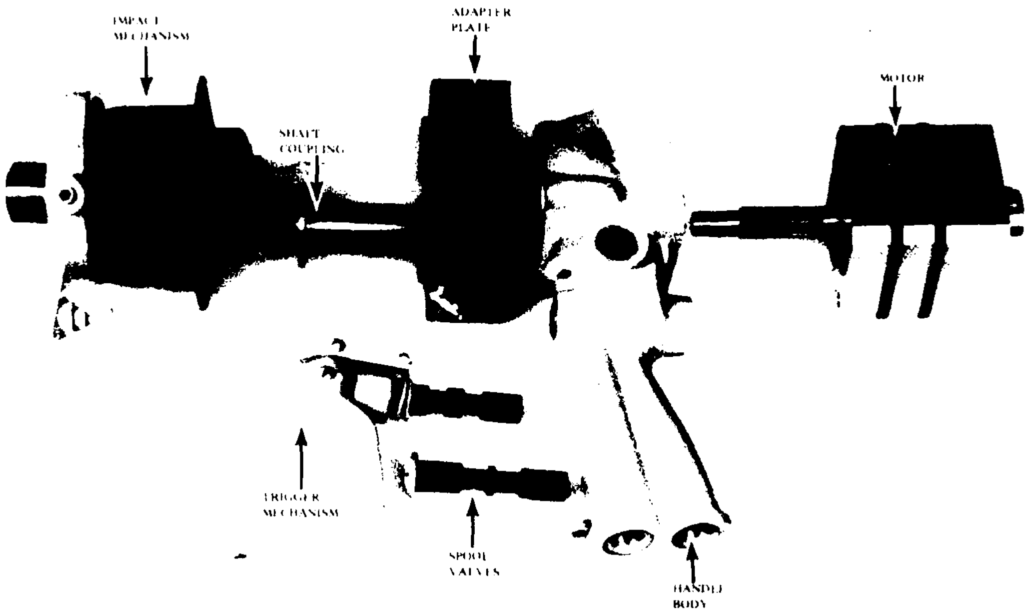


Figure 19. Major components of impact wrench include (from left to right): impact mechanism, shaft coupling, adapter plate, pistol-grip handle, and motor.

An adapter plate was fabricated to accommodate mounting of the impact mechanism on the front of the handle. The adapter plate was made long enough to provide room for installation of a flexible coupling between the rotor and impactor. Since the flexible coupling was not needed, a rigid shaft coupling was fabricated from AMPCO 45* (Figure 19). This material was used because of its excellent corrosion and wear resistant properties, high strength, extra toughness, and bearing qualities. The extra strength and toughness properties are needed to withstand the shock loads from the mechanism during impact; the bearing properties are needed to permit the rotor spline to slide axially within the coupling, and the coupling to slide within the impactor. The flange on the coupling mates with the Torlon thrust washer mounted in the handle adapter plate. This bearing serves as the support for the motor shaft coupling. Radial grooves on the bearing provide a path for the seawater exiting the forward motor bearing to cool and lubricate the bearing.

Propeller Cleaning Brush

The propeller cleaning brush was also designed to provide at a minimal cost a representative tool suitable for laboratory and field evaluation by technical personnel; therefore, some components of the seawater-powered impact wrench were adapted and modified as required.

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*An extruded-aluminum-nickel bronze material.

The propeller cleaning brush (Figure 20) weighs 12 pounds in the air, 8.5 pounds submerged, and is 11 inches long. The unidirectional motor is attached to the back of the tool with four high strength steel bolts. The forward end plate of the motor has O-ring seals that direct pressurized fluid from the tool handle. Dowel pins were placed between the motor and the handle to ensure proper alignment of the motor shaft with the output brush shaft.

The pistol-grip handle used for the cleaning brush is the same handle used for the impact wrench. This handle was chosen because of its availability and low cost. The handle is cast aluminum and uses standard hydraulic spool valves for flow control and tool reversing.

While long operational life was not a goal for the handle, high velocity seawater flowing through the intricately machined passages of the handle caused the aluminum to erode. To estimate handle life, seawater was passed through the handle at flow and pressure conditions for 10 consecutive hours. An inspection of the handle showed moderate erosion (see Figure 21). Handle life was estimated to be about 50 hours, which was deemed acceptable for the intended application.

The handle flow tests and tests with the impact wrench also revealed unacceptable leakage from the two spool valves; leakage rates as high as 15% were measured. To reduce the leakage for the propeller brush, the standard reversing spool was replaced by a nonsliding design with positive sealing O-rings at the spool lands. Also, by eliminating the return flow from the motor, the return port in the handle could be plugged, effectively reducing that leakage path.

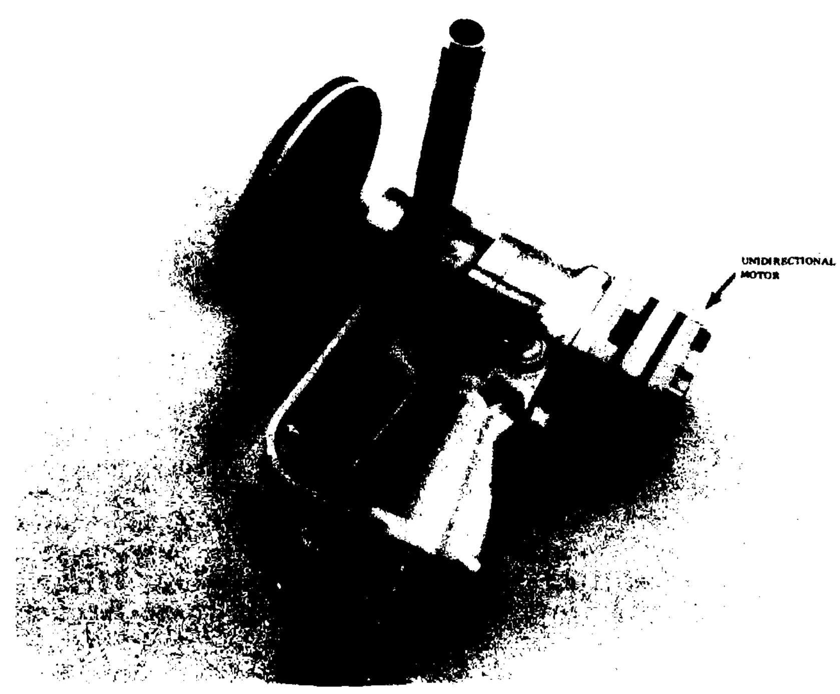


Figure 20. Propeller cleaning brush with unidirectional motor.

Figure 21. Erosion of pistol-grip handle after handle flow test. Flow erosion was noted in the spool bores as well as on the adapter back plate.

Another major component of the cleaning brush is the nose piece. It consists of the output shaft, diver-assist handle, and barrel. The nose piece was designed and fabricated at NCEL.

The output shaft is made from 304 stainless steel and is supported by two Rulon "J" combination sleeve and thrust bearings. These bearings were designed to be operated immersed in seawater.

The assist handle and barrel are 6061-T6 aluminum. The assist handle can be locked around the barrel in several radial positions, depending on the diver's preference. The barrel is free flooded and the front motor bearing lubricant flow passes through the barrel, giving a small continuous purging water flow.

Seawater Hydraulic Power Source

The seawater hydraulic power source (SWHPS) (Figure 22) is a portable, self-contained means of delivering pressurized seawater to diver tools. The SWHPS design was based on the operational requirements for Navy construction diver operations. In operations the SWHPS may be located on the diving support platform, on shore, on a pier, or at wharfside. To operate in all these situations, the SWHPS must be portable, able to draw seawater from the sea continually, filter the seawater, and provide sufficient pressurized flow to power two tools at one time.

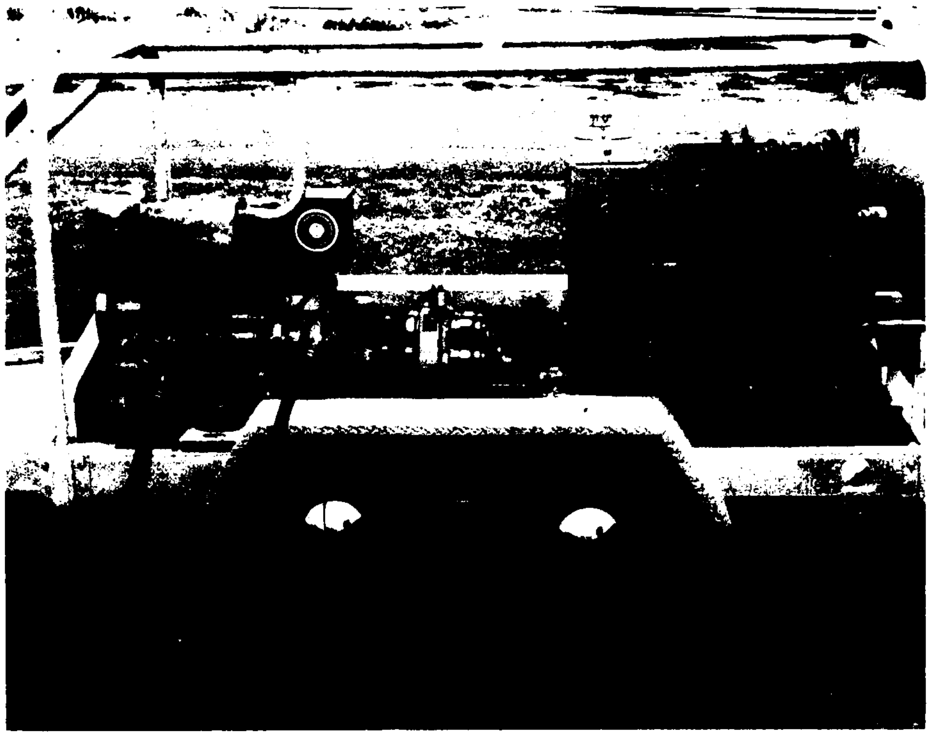


Figure 22. Portable seawater hydraulic power source. Hose reel with 250 feet of hose shown to right. Main seawater pump is in center of photograph.

The SWHPS is capable of delivering flow rates up to 12 gpm at pressures to 2,000 psi, and is mounted on a trailer for portability. All components of the SWHPS were designed to be resistant to the corrosive effects of the marine environment. The SWHPS is 192 inches long, 80 inches wide, and 70 inches high and weighs 3,500 pounds. Though the size and weight of the SWHPS were not ideal, ease of access to system components during the experimental test and evaluation took precedence over small size and light weight. Future models will be considerably smaller and lighter.

The SWHPS is diesel-powered; a system schematic is given in Figure 23. A 35 hp, three-cylinder, air-cooled diesel engine drives two pumps, a belt-driven suction pump, and a direct-driven main pump. During operation, the centrifugal suction pump draws seawater from the sea and is capable of operating against a 15-foot suction head. A strainer and a foot valve are mounted on the bottom of the 2-1/2-inch suction hose - the strainer keeping large particles from entering the system and the foot valve maintaining system prime when the pump is not operating.

The suction pump forces seawater through the main filter system to fill the reservoir and provides coolant to one side of a counter-flow heat exchanger. A flow control valve limits the outlet flow of the suction pump.

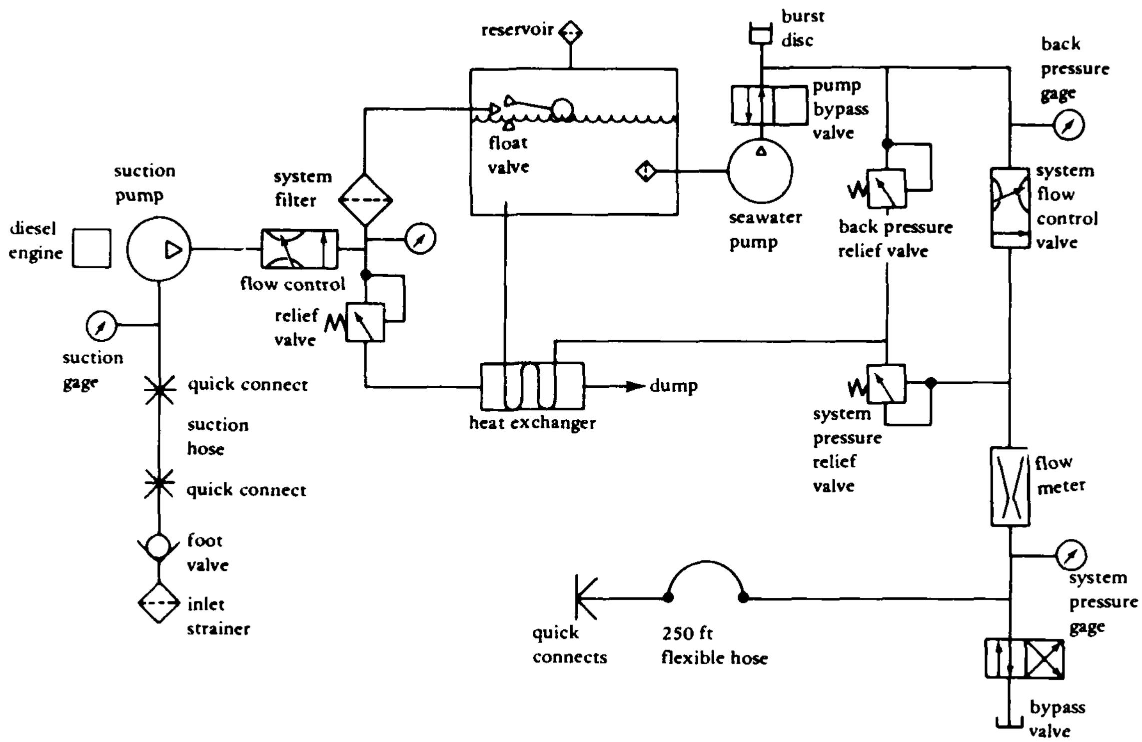


Figure 23. Flow schematic for the portable seawater hydraulic power source.

The filtration system was sized to permit continuous operation in heavily silted waters for up to 8 hours and is capable of removing particles down to 10-micron nominal size. The multi-pass heat exchanger provides cooling for by-passed seawater from the main hydraulic system controls. The cooled fluid is returned to the reservoir instead of being dumped overboard. With this system the quantity of fresh seawater entering the reservoir is minimized and the filter load reduced.

A 50-gallon stainless steel reservoir provides 14 to 16 inches of positive head to supply the main seawater pump. The reservoir is equipped with: (1) an access panel for cleanout; (2) a float valve, which shuts off suction pump flow when the reservoir is full; (3) a sight gauge for inspection of fluid level; and (4) baffles to reduce fluid aeration and promote settling of sediments.

Filtered seawater from the reservoir is directed to the inlet manifold of the main hydraulic pump. This eight-cylinder, dual-fluid radial piston pump (Figure 24) is driven from the diesel crank shaft via a 2:1 epicyclic gear reducer. Each of the eight internal pistons pump oil which is directed to an adjacent tubular diaphragm. Compressing the cylindrical diaphragm forces the seawater out, and relaxing the diaphragm brings the seawater in. Ball-type check valves on each end of the diaphragm control the fluid direction. Seawater from each diaphragm is directed to a central distribution manifold and then to the SWPHS' control circuit.

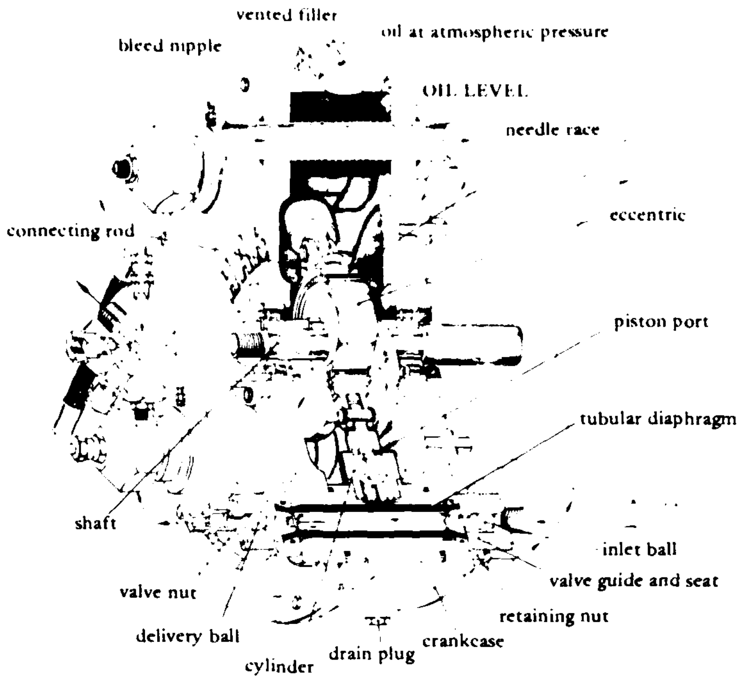


Figure 24. Main seawater hydraulic pump showing dual fluid pumping arrangement.

Control circuitry is provided so that the SWHPS can be operated in either a closed- or open-centered mode. An adjustable, pressure-compensated flow control valve limits the seawater flow to the diver tool. Excess flow from the pump is directed through the heat exchanger from the back pressure relief valve. An adjustable pressure relief valve located in the supply line limits system pressure to the tool.

The SWHPS is equipped with gauges for monitoring system pressure, delivered flow and pressure, and reservoir temperature. A pressure gauge is provided for monitoring the performance of the suction system and condition of the filters. Additionally, a 250 foot 1/2-inch ID synthetic hose is mounted on a hand-powered hose reel.

TEST AND EVALUATION

Test Arrangement

For evaluation of the seawater hydraulic tool system, components of the impact wrench and propeller cleaning brush were laboratory bench-tested; then the assembled system tank-tested with divers. The reversible motor, mechanically coupled to the impact mechanism and attached to the tool handle, was evaluated by itself. The unidirectional motor was also tested by itself and attached to the propeller cleaning brush handle.

The laboratory arrangement for the motor and tool component evaluations is shown in Figure 25. A schematic for the test bench is shown in Figure 26. Motor and motor-handle tests were conducted with the motor coupled to an oil hydraulic load pump via a rotary torque transducer. The load was applied by throttling the outlet of the pump using a needle valve. Tests were run using fresh tap water and fresh seawater pumped from Port Hueneme Harbor. Flow rate, inlet pressure, temperature, rotary torque, and revolutions per minute were electronically monitored. The data were automatically recorded at regular intervals (usually once per minute) and processed by a microcomputer, which was programmed to calculate input and output power and operational efficiencies. The data were recorded on a line printer. Each data point represents the average of five samples.

The arrangement for the motor-impactor tests is in the section describing the impact wrench tests. The motor was directly coupled to the impact mechanism by a short adapter shaft. A bolt torque tester using a standard impact socket measured output of the impact mechanism. Tightening of the bolt resulted in compression of a hydraulic cylinder. The relationship between cylinder pressure and bolt torque was obtained by using an accurate torque wrench to tighten the bolt to specific values and then recording the pressure. The pressure of the oil in the tension-cylinder and seawater inlet flow and pressure to the motor were continuously monitored and recorded during the tests.

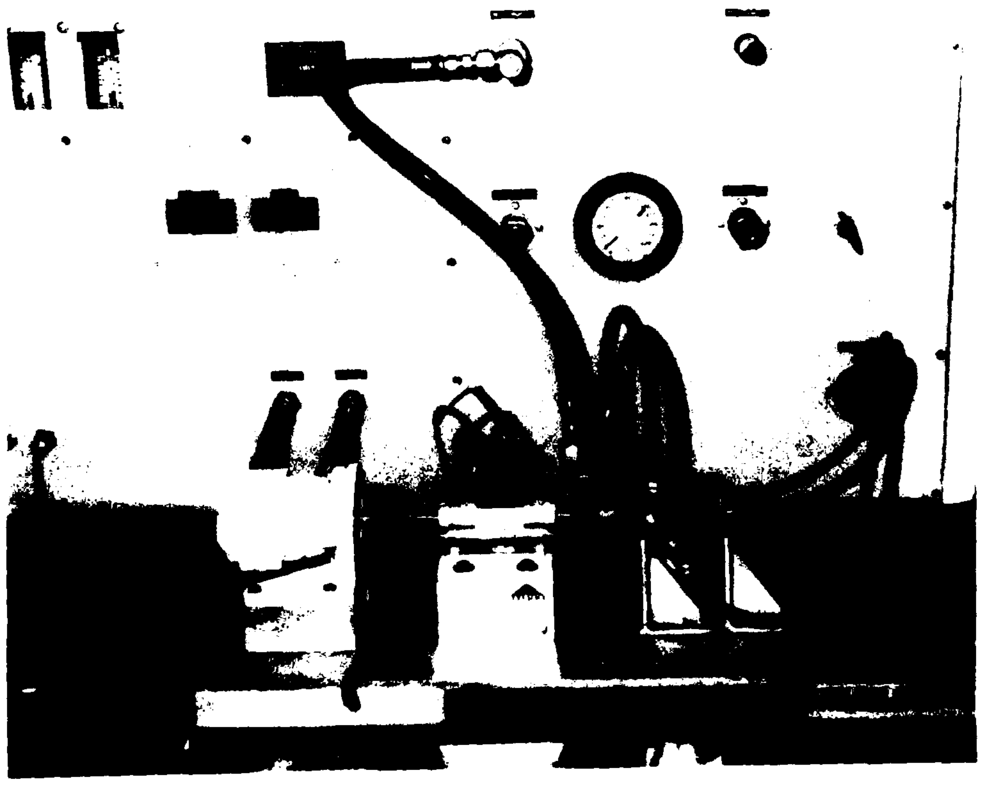


Figure 25. Laboratory test arrangement for the motor and tool components evaluation.

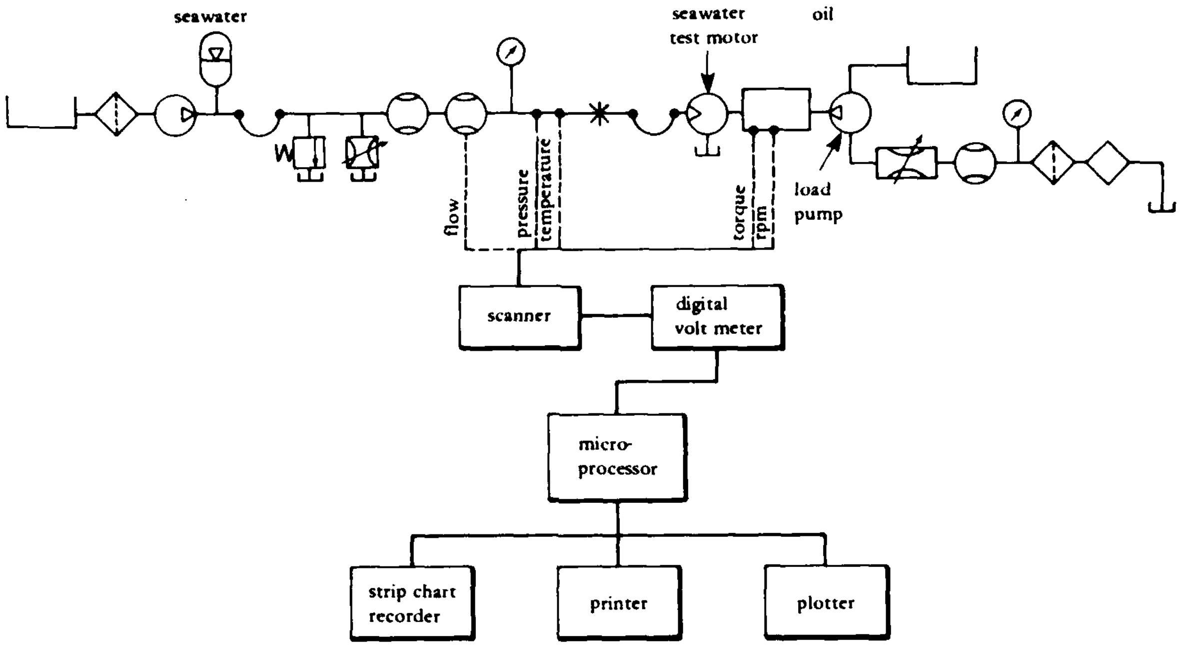


Figure 26. Test bench flow and instrumentation schematic.

Tests of the complete experimental tool system were conducted at the NCEL shallow water test facility using Navy divers. The test facility consists of a 30-foot-diam tank filled with seawater to a 12-foot depth. The divers used the impact wrench to tighten nuts and bolts and drill holes in steel plate. The propeller cleaning brush was used on the side of the tank.

Motor Evaluations

Improved Reversible Motor. Performance tests on the improved reversible motor were conducted over a period of 35 days for a total running time of 205 hours. The motor was operated at 2.3-hp output (80% of rated output) for most of the test period. Data were recorded at one 1-minute intervals over the duration of the test. Each day the motor was tested to over 3-hp output and on several occasions power output was increased to over 4 hp. Performance data were obtained daily at seawater flow rates of 5, 6, and 7 gpm at pressures up to 1,200 psi.

The motor performed exceptionally well and met the design objectives of providing 3-hp output with 70% overall efficiency at a seawater inlet pressure of at least 1,000 psi. The overall performance of the motor during the 200-hour test is shown in the plot of mechanical and volumetric efficiency in Figure 27. Data plotted represents performance of the motor with an input flow of 6 gpm and a load pressure of 950 psi.

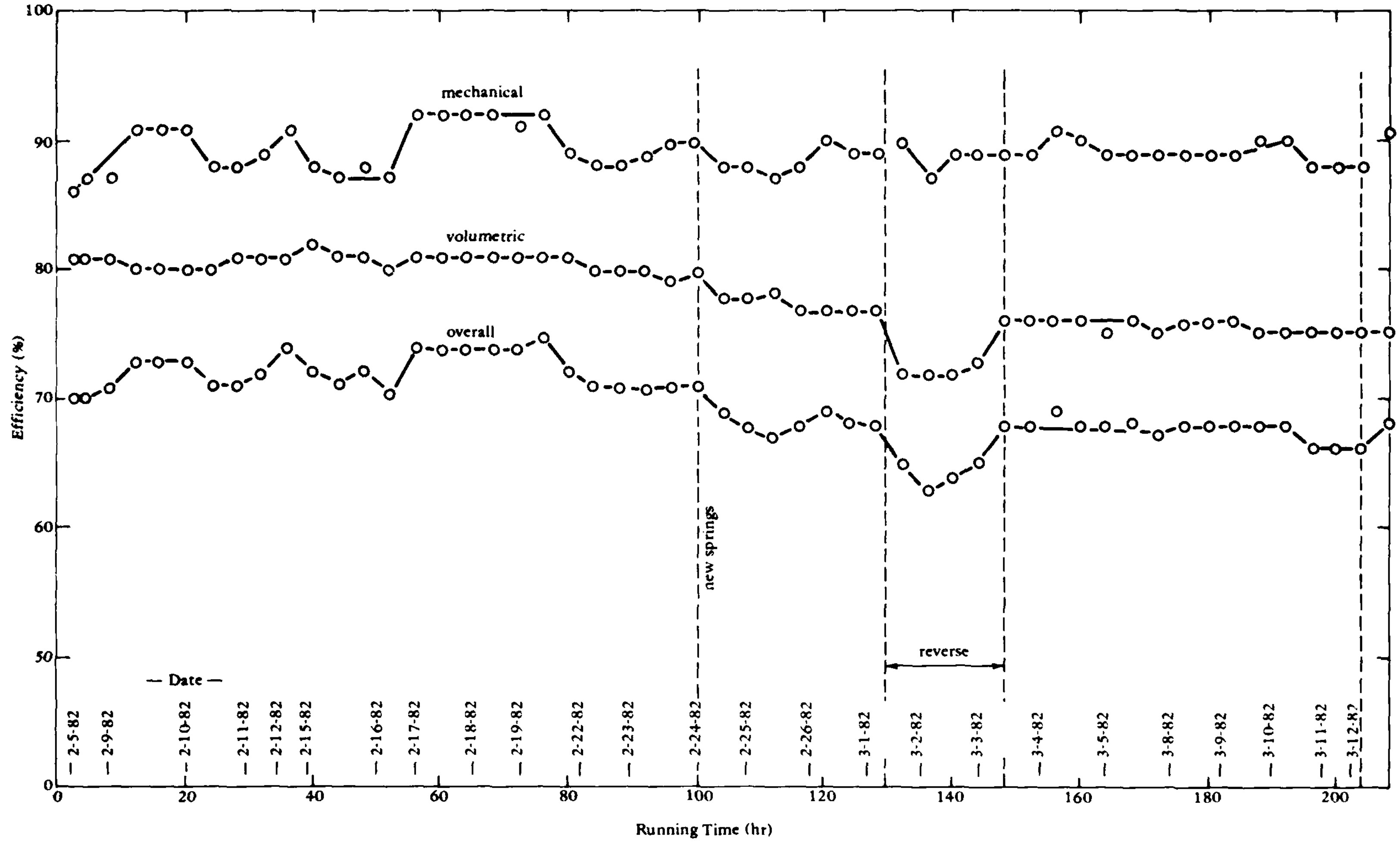


Figure 27. Vane motor efficiency during 200-hour endurance test. Drop in efficiency at 130 hours was attributed to cross port leakage on back ports due to faulty O-ring groove.

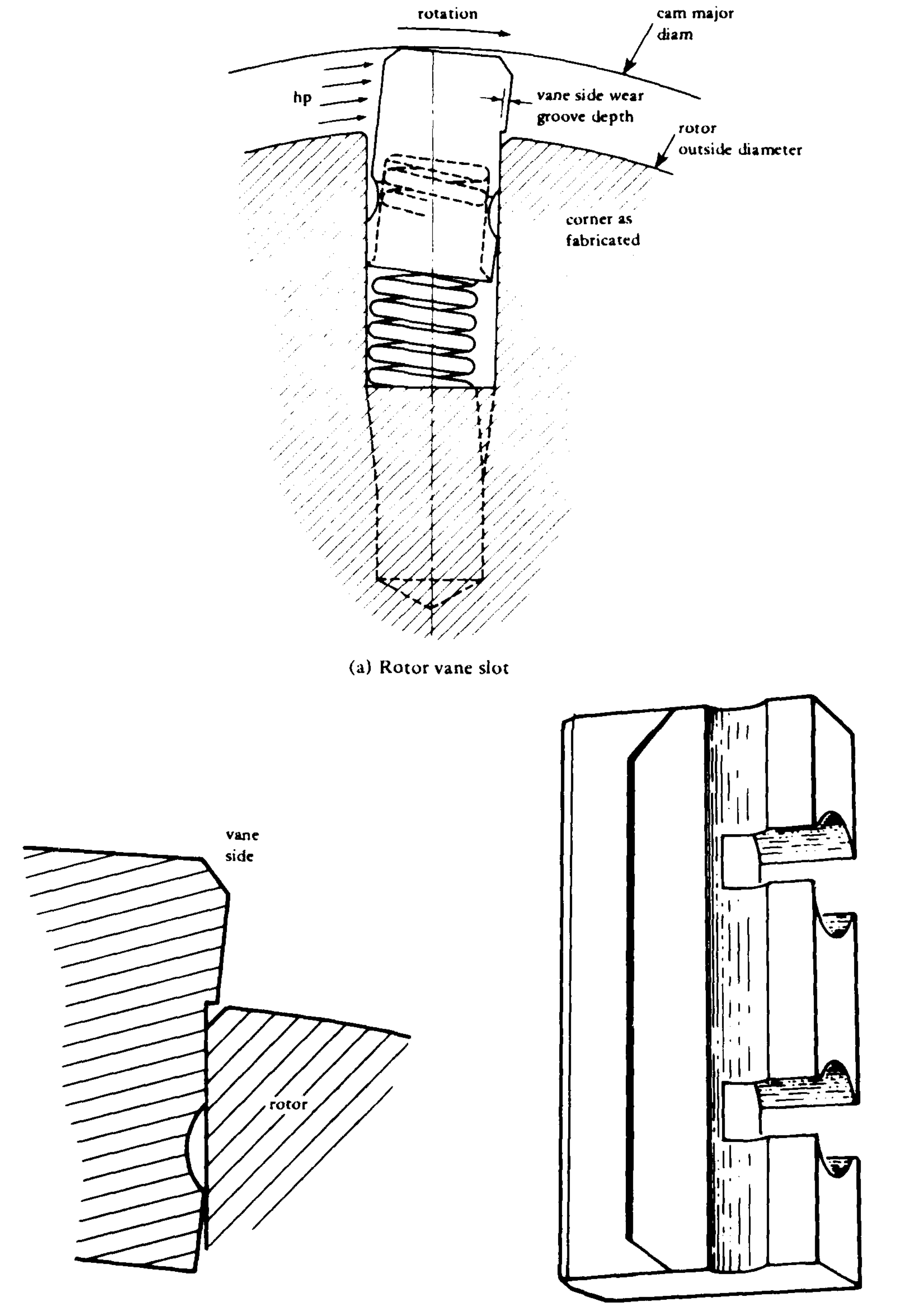
During the first few hours of testing, motor efficiency increased with wear-in of the vanes. Initially, the contact between the vane tip and the cam is only a line and does not produce a good seal. As the vane wears in, the line contact becomes a surface that conforms to the cam's profile and reduces the tip leakage.

Over the life of the endurance test, volumetric efficiency decreased from 80 to 75%. The decrease was attributed to wear on the bearings, side plates, and the face of the vane where it contacts the outer edge of the rotor. On the motor tested, the edge of the rotor had been cut on a 45-degree bevel as shown in Figure 28. Fabrication drawings called for this edge to be on a radius. The continuously reciprocating action of the vane on the sharp edge cut the plastic vane. The wear on the face of the vane caused excessive clearance between the vane and vane slot which resulted in a higher leakage rate. Performance curves for the motor after 200 hours of operation are given in Figures 29 and 30. Data for the curves are provided in Table 1. Note that the required operational performance goal of 70% overall operational efficiency was achieved with 7-gpm flow and a 1,093-psi load.

Mechanical (torque) and volumetric (leakage) efficiency of the motor as related to shaft speed and operational pressure are shown in Figure 31 (a and b). It was expected that mechanical efficiency might decrease slightly with increased shaft speed due to increased windage losses, but this did not occur indicating that windage losses were not significant. Mechanical efficiency did increase with increasing pressure, indicating that frictional losses stayed relatively constant and became less significant as overall motor torque output increased. As was expected, volumetric efficiency decreased significantly with increased pressure, but increased with increased flow rate (rpm).

The motor was disassembled after 100 and 200 hours when it was suspected that one or more springs had broken. A broken or weak spring is characterized by a knocking sound at pressures over 900 psi. The knocking results from the vane bouncing off the track. Vane bounce is caused by an imbalance of forces between the tip and bottom of the vane. As the spring force weakens, the pressure-generated tip forces exceed forces on the bottom of the vane, and the vane leaves the track. The pressure behind the vane is then relieved, and the vane slams back against the track. At 100 hours, one spring was broken, and others were so severely worn that all springs were replaced. As discussed earlier the wear resulted from the spring contact with the rotor as shown in Figure 7. The spring failure was not due to fatigue failure, as had occurred at 50 hours in previous tests with the experimental motor. At 200 hours of operation no broken springs were found, but again the wear was severe. Performance of the motor below the knock point was not noticeably affected by the weak springs. However, continued running with the vanes knocking would probably have resulted in severe damage to the vanes, vane track, and side plates.

Inspection of the vanes following 100 and 200 hours of operation showed very little tip wear. The vane height did not measurably change in 200 hours of operation. When the expected dimensional changes due to moisture absorption and subsequent swelling are considered, vane tip wear was approximated at 2.5×10^{-4} in./100 hr. As discussed earlier, wear on the face of the vane opposite the pressure forces where the vane contacted the sharp edge of the rotor was excessive and was measured at 1.5×10^{-3} in./100 hr.



(a) Rotor vane slot

(b) Rotor corner was beveled, resulting in high side wear on vane

(c) Side wear resulted from rubbing contact with rotor on face opposite pressurized face as shown in (a)

Figure 28. Wear interaction between vane and rotor.

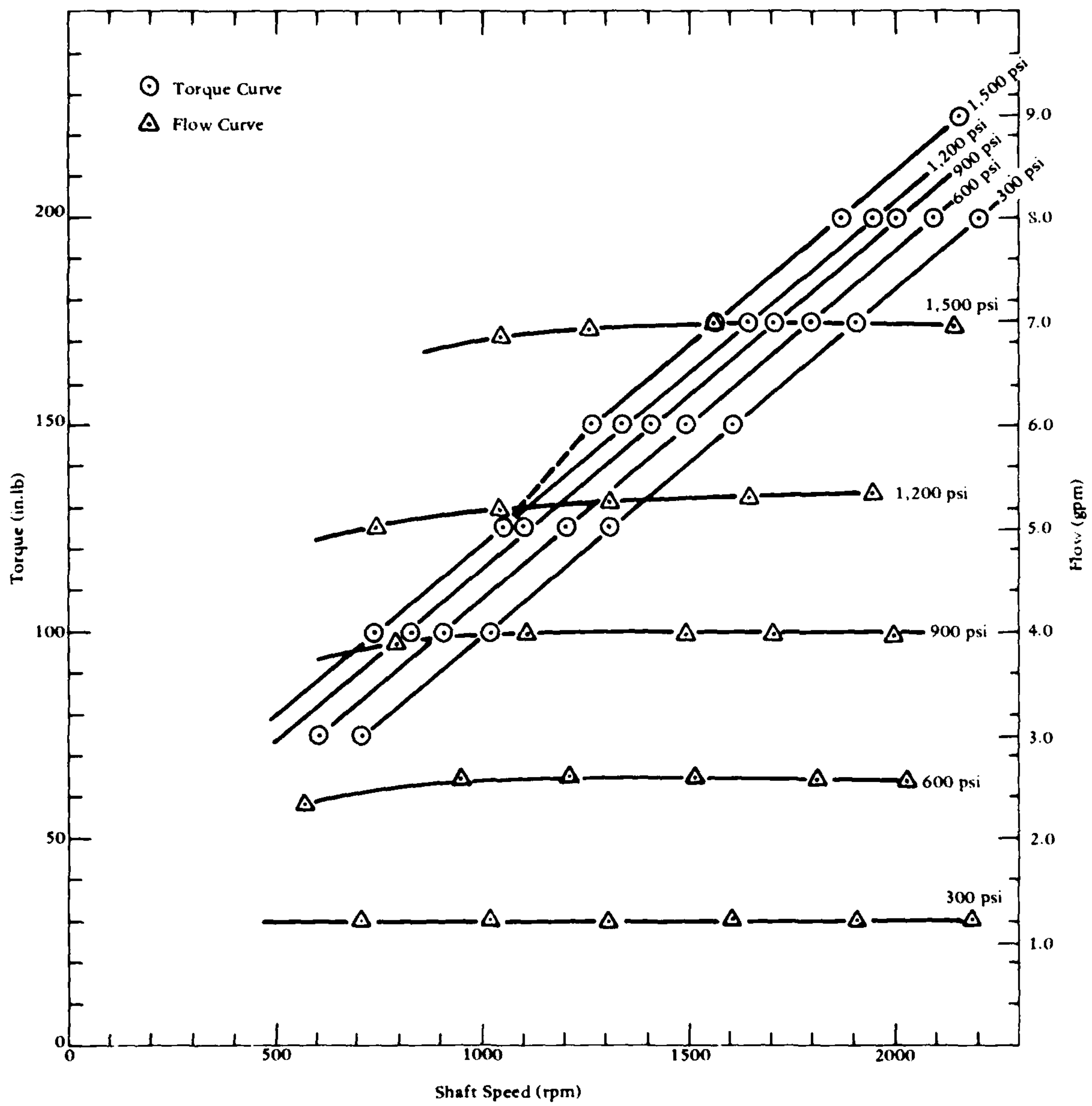


Figure 29. Performance curves for reversible motor after 200 hours of operation.

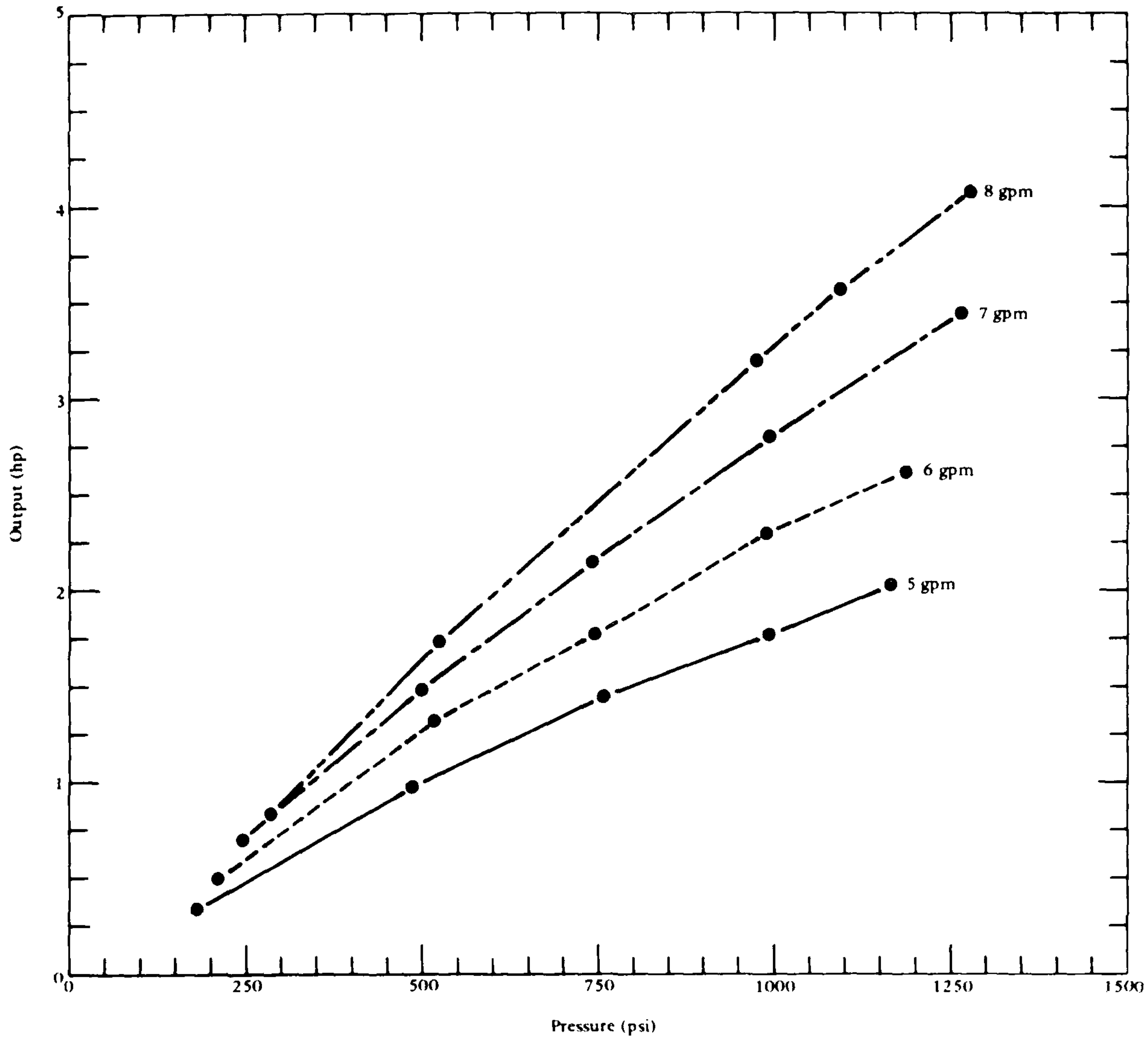


Figure 30. Reversible seawater motor endurance test results showing power output.

Table 1. Reversible Motor Performance Data
After 200 Hours of Testing

Flow (gpm)	Pressure (psi)	Torque (in.-lb)	Shaft Speed (rpm)	Input (hp)	Output (hp)	η_M [a]	η_V [b]	η_T [c]
5.01	176	17	1,298	0.52	0.35	0.78	0.88	0.69
5.02	512	56	1,135	1.50	1.00	0.87	0.77	0.67
4.99	751	85	1,079	2.19	1.45	0.90	0.74	0.66
5.01	991	110	1,025	2.90	1.79	0.89	0.70	0.62
5.00	1,224	136	997	3.57	2.16	0.89	0.68	0.60
6.00	202	22	1,590	0.71	0.55	0.85	0.90	0.77
5.99	515	56	1,420	1.80	1.26	0.87	0.81	0.70
6.00	735	83	1,359	2.57	1.79	0.90	0.77	0.69
6.01	1,006	114	1,304	3.53	2.36	0.91	0.74	0.67
6.01	1,252	140	1,239	4.39	2.76	0.90	0.70	0.63
7.00	241	23	1,856	0.98	0.68	0.77	0.90	0.69
7.00	496	54	1,719	2.03	1.48	0.87	0.84	0.73
7.01	755	84	1,648	3.09	2.20	0.89	0.80	0.71
6.99	997	112	1,579	4.07	2.81	0.90	0.77	0.69
6.99	1,222	139	1,533	4.99	3.37	0.91	0.75	0.68
8.00	279	26	2.089	1.30	0.86	0.74	0.89	0.66
7.99	493	51	1,992	2.30	1.60	0.82	0.85	0.70
8.01	747	81	1,918	3.49	2.46	0.86	0.81	0.70
7.99	1,010	113	1,868	4.71	3.35	0.89	0.79	0.71
7.99	1,187	134	1,827	5.54	3.88	0.90	0.78	0.70

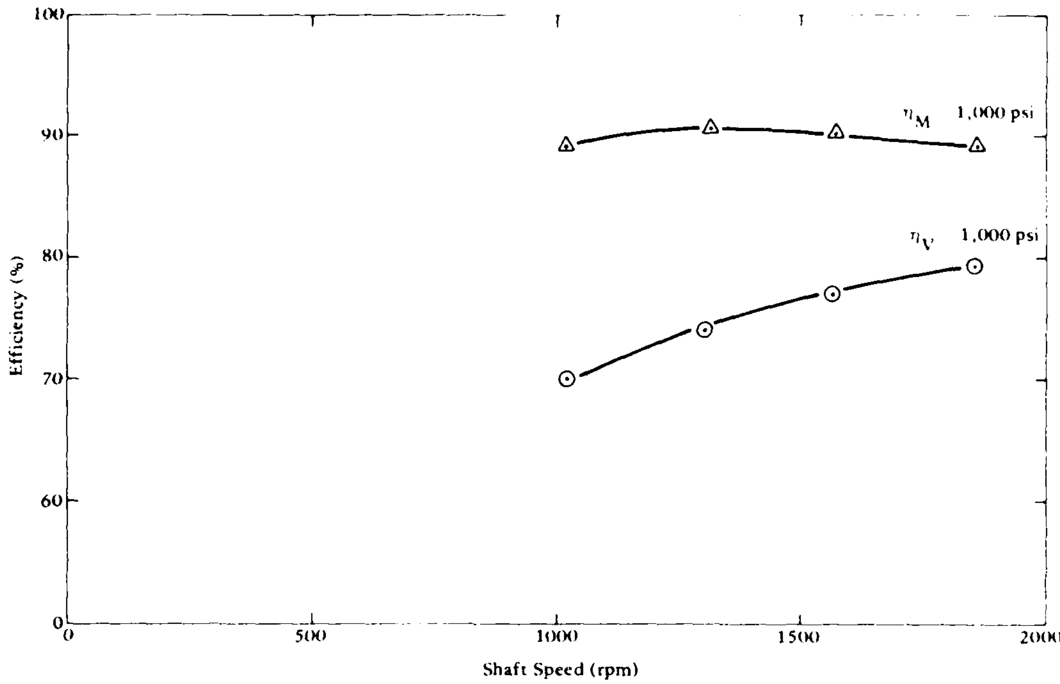
[a] η_M = Mechanical Efficiency (friction).

[b] η_V = Volumetric Efficiency (volumetric).

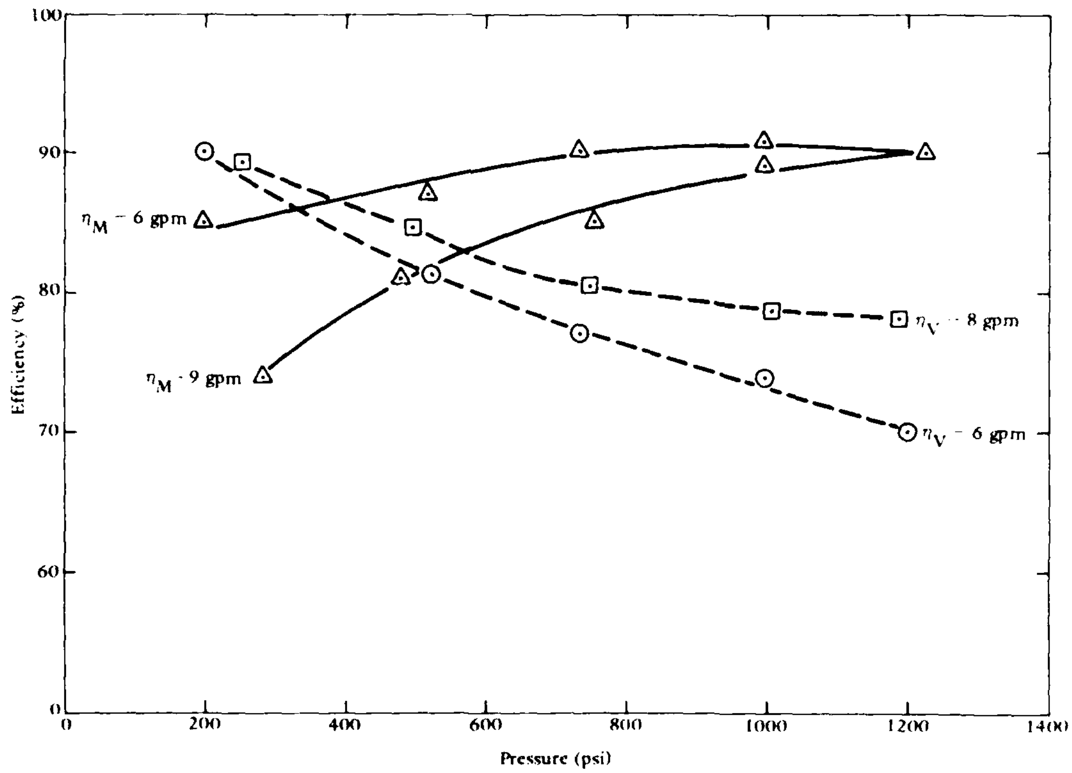
[c] η_T = Overall Efficiency (hp in/hp out).

No significant wear was noted on either of the Torlon side plates or the vane track surface. No wear was noted on the rotor. The front and back bearings were heavily scored in several places. The wear location was such that minor misalignment was suspected. The bearings had not been line-bored following installation, which could account for the misalignment. Measurement of the bearings following the tests showed a slight out-of-round wear (2×10^{-4} inches).

The motor was run in the reverse direction for approximately 20 hours during the 200-hour test. As shown in Figure 27, volumetric efficiency dropped approximately 5% in the reverse direction, and bearing leakage flow doubled. Bearing flow is primarily supplied by the flow restrictors. At first, the high-bearing flow was attributed to a faulty restrictor; but later, following disassembly, it was noticed that a leakage path had occurred between the end plate and the flexible side plate at the reverse direction high-pressure ports (see Figure 32) because of an ineffective O-ring seal. Inspection and measurement showed that the groove was too deep, resulting in insufficient squeeze of the O-ring.



(a) Mechanical and volumetric efficiency of reversible motor as related to shaft speed (rpm). As expected, volumetric efficiency increases with speed.



(b) Volumetric and mechanical efficiency of the reversible vane motor at different operational pressures and flow rates.

Figure 31. Characteristics performance curves for reversible vane motor.

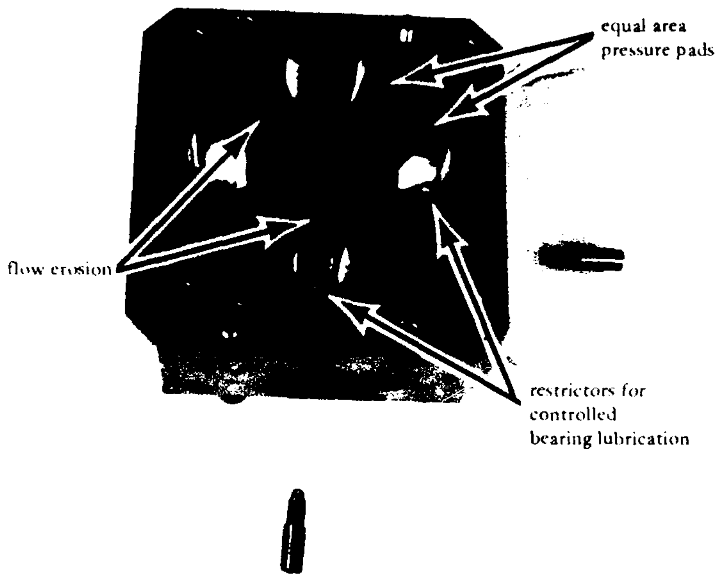


Figure 32. Aft housing pressure pad areas showing flow erosion areas. Leakage resulted during reverse direction operation because of insufficient O-ring squeeze.

The primary function of the flow restrictors is to provide bearing lubrication in both the forward and reverse directions. The present technique uses commercially available restrictors that are fabricated from grade 303 stainless steel. Both forward and reverse restrictors functioned properly during the 200-hour test and 35-day immersion. In a separate test, another flow restrictor caused bearing failure. This failure was attributed to severe internal corrosion which resulted from long-term (9 months) exposure to seawater.

Motor Materials Evaluation. Performance tests using the experimental motor as the test bed were conducted using different materials for the vanes and vane track. The materials evaluated are listed in Table 2. These tests were done to identify materials that would reduce leakage, friction, and wear on the critical motor components. Fresh tap water was used as the motor working fluid. During each test, the motor was operated with 6-gpm, 1,100-psi input, and data were recorded at 5-minute intervals. After running for a period of 5 to 8 hours, the motor was disassembled and inspected for wear. Vane, cam, and rotor dimensions were measured. Vane tip wear was used as the primary indicator of material performance. To minimize the effect from water absorption dimensional changes, the vanes were soaked for at least 48 hours before they were tested in the motor. Measurements of the radial length, axial height, and thickness of the vanes were taken before the test and at times when the motor was disassembled for inspection. The tests were concluded if vane tip or other component wear were found to be excessive.

Table 2. Materials Evaluated During the Motor Performance Tests

Material[a]	Composition	Component
Torlon		
4203	0.03% TiO_2	vane
4275	20% power graphite, 3% PTFE	vane
4301	12% power graphite, 3% PTFE	vane
4347	12% power graphite, 8% PTFE	vane
7130	30% fiber graphite, 1% PTFE	vane
Alumina Ceramic, SC-98D	98% Al_2O_3 plus proprietary wear additives	cam
Inconel 625	high Ni-Co based alloy	cam

[a]Properties of materials are listed in the Appendix.

Data from the vane material tests are presented in Table 3. Five different grades of Torlon were evaluated; two grades of Torlon were tested with a ceramic cam. Wear rate is presented for the first few hours of each test and in two cases for extended periods of operation. The initial wear rate was higher than the extended wear rate. This was expected: the initial contact area of the vane tip with the cam is small and increases as wear-in occurs. The small contact area results in high PV loading on the vane tip and consequently high wear. As the vane wears in, the contact area increases, PV loading decreases, and the tip wear rate decreases.

Of the different Torlon materials tested, 4347, 4301, and 4275 showed lower tip wear than other materials. The second test with 4347 showed considerably greater wear than in the first test. Also, the first test with 4275 showed considerably greater wear than in the second test. In both high wear conditions, the cam surface, side plates, and vane edges showed heavy scoring and wear. The vane tips, vane edges, and side plates also contained large quantities of Inconel particles embedded in the plastic. Though the origin of these particles is not yet known, several possibilities exist: (1) the metal was generated from the springs rubbing against the rotor, (2) excessive PV loads at the vane tip could have removed metal from the cam, (3) excessive flexibility of the side plates could have caused high-side stress on the rotor and removal of metal, or (4) contaminants from outside the motor could have removed metal at the tip or side plates. Regardless of the cause, the result appears to be a migration of metal to the vane tips and embedding in the plastic vane. The metal-to-metal contact causes heavy

scoring of the cam surface and scoring and wear on the vane tips. This same condition was noted in other tests. Tests to determine the location of where the metal is generated are planned for the near future.

A comparison of the lowest wear conditions from each of the Torlon materials tested showed some interesting trends. The data from test 1 and 7, with 3 and 8% PTFE* shows that lower wear results from an increase in PTFE content. A comparison of test 1 with test 5 shows an increase in wear with an increase in graphite powder content. As expected, the test of the Torlon resin with no lubricating fillers showed high wear. This material was designed primarily for electrical and structural applications. The addition of graphite fibers provided a significant increase in the mechanical strength but a marked decrease in the wear over the resin with no additives.

A cam was fabricated from an alumina ceramic material that contained 98% alumina ceramic and proprietary wear additives. This material was selected based on the manufacturer's recommendations for the wear conditions. The cam was fabricated by the ceramic manufacturer to the same dimensions, surface finish, and profile of the standard Inconel cam.

Two of the low-wear Torlon materials were tested with the ceramic cam. As shown in Table 3, tests 6 and 9, both Torlon materials exhibited high wear rates with little difference between the materials tested. It appeared from visual examination of the vane tips that the wear was uniform and similar to that which might be experienced with surface polishing operations. No damage or wear was noted on the ceramic surface, and little or no metal was noted on the vane tips. The cam surface was light grey in color following the tests which indicated that the Torlon resin did rub off onto the cam.

Side plates were fabricated from a metalized carbon material (Metcar grade M271), which consists of a carbon-graphite base material impregnated with a copper-lead alloy. Previous tests with this material showed that it is compatible with both Inconel and Torlon. One of the main reasons for the Metcar test was to obtain data on motor performance with nonflexible side plates. If the flexible side plates significantly improved sealing at the side of the rotor, then the results with nonflexible plates should show lower volumetric efficiency.

Data from the Metcar tests are presented in Table 4. Little or no difference was noted between the flexible Torlon side plates and the stiff Metcar side plates. In general, the operation with Torlon side plates was only slightly more efficient than with the Metcar side plates. The starting torque for the motor with the Metcar side plates was significantly less than for the Torlon side plates. This was attributed to lower static friction of the side plates on the side of the rotor due to the fact that the plates do not flex against the rotor with pressure. After the motor was disassembled, it was noted that the sides of the Inconel rotor were copper-colored, indicating a transfer of material from the Metcar side plates.

*Polytetrafluoroethylene.

Table 5. Vane Materials Test Results

Test No.	Vendor	Tip wear 10^{-3} (in./hr)	Test (hr)	Performance (6 gpm 1,100 psi) η_M[a]	η_V[b]	Material Content Graphite ( )	PTFE ( )	Other	Comments
1	4301	0.67	7	0.91	0.89	12 powder	8	--	Metal in tip; light scratches; deep grooves; heavy side wear (7×10^{-3} in./15 hrs); 50 area wear; light scoring on cam
2	7130	0.26	8	0.79	0.73	30 fiber	--	--	Heavy metal in vane tip; light wear groove; heavily scored tip wear; cam heavily scored; heavy wear on side plates
3	4703	0.66	8	0.79	0.81	--	--	TiO_2 3	Even, heavy wear; heavy side groove wear (5×10^{-3} in.); fine score marks on cam
4	4275-1	0.23	8	0.78	0.82	20 powder	3	--	Heavy tip wear; much metal in tip; cam scored evenly
5	4275-	0.11 0.07	9 16	0.79	0.64	20 powder	3	--	Slight metal in tips; vane tips polished; heavy wear on side groove (14×10^{-3} in./16 hr); cam surface good
6	4275-C[c]	2.4	61	0.79	0.76	20 powder	3	--	Polished tip; 100 area wear; color on cam; side wear minimal
7	4347-1	0.04 0.019	81 30	0.90	0.88	12 powder	8	--	Light wear on tip; 75 tip area wear; no metal; side wear (6×10^{-3} in.); voids in vane material; one broken vane; cam excellent
8	4347-2	0.16	5	0.79	0.73	12 powder	8	--	Very little metal in tips; medium wear on side groove; heavy side plate wear; used new rotor; cam heavily scored
9	4347-C[c]	2.3	6	0.77	0.78	12 powder	8	--	Polished tip; no metal

[a] η_M = Mechanical Efficiency (friction).

[b] η_V = Volumetric Efficiency (volumetric).

[c] Ceramic cam was used.

Table 4. Metcar and Torlon 4301 Side Plates Test Results

[7-gpm flow rate was used.]

System Pressure (psi)	Material	Torque (in./lbs)	Speed (rpm)	Overall Efficiency (%)
600	Metcar	69	1,675	75
	Torlon 4301	68	1,650	73
900	Metcar	108	1,560	73
	Torlon 4301	108	1,600	75
1,200	Metcar	144	1,490	70
	Torlon 4301	142	1,575	72
1,500	Metcar	180	1,410	66
	Torlon 4301	176	1,490	68

Additional tests were planned to substantiate the findings of the Metcar side plates, but during the test one side plate was broken and the test could not be performed. Further tests will be conducted at a future date.

Motor Modification Tests. Minor modifications made to the improved vane motor were evaluated to determine the effect on motor performance: (1) sensitivity of the motor to certain design factors for improving motor performance by increasing operational efficiency; (2) reduction of the occurrence of vane bounce; and (3) improvement in the start-stop characteristics of the motor.

Modifications evaluated included changing the sweep angle of the inlet and outlet ports, adding lead grooves to the inlet ports, changing the spring hole shape, and reducing side plate flexibility. The modifications were evaluated by running the motor against a load and determining operational characteristics.

The normal sweep angle on the inlet and outlet ports, which are located on the flexible side plates, is 40 degrees. Side plates were fabricated with sweep angles of 35 and 45 degrees. In addition, the side plates with a 35-degree sweep angle were modified to include a 5-degree V-groove filed in the leading edge of the port (Figure 33).

Figure 34 shows the result of port sweep angle on operational efficiency of the motor. As might be expected, volumetric efficiency (leakage) was affected most, with a 7% drop from 35 to 45 degrees. Mechanical efficiency was only slightly affected by the sweep angle. Adding the V-groove to the 35-degree ports slightly reduced both volumetric and mechanical efficiency.

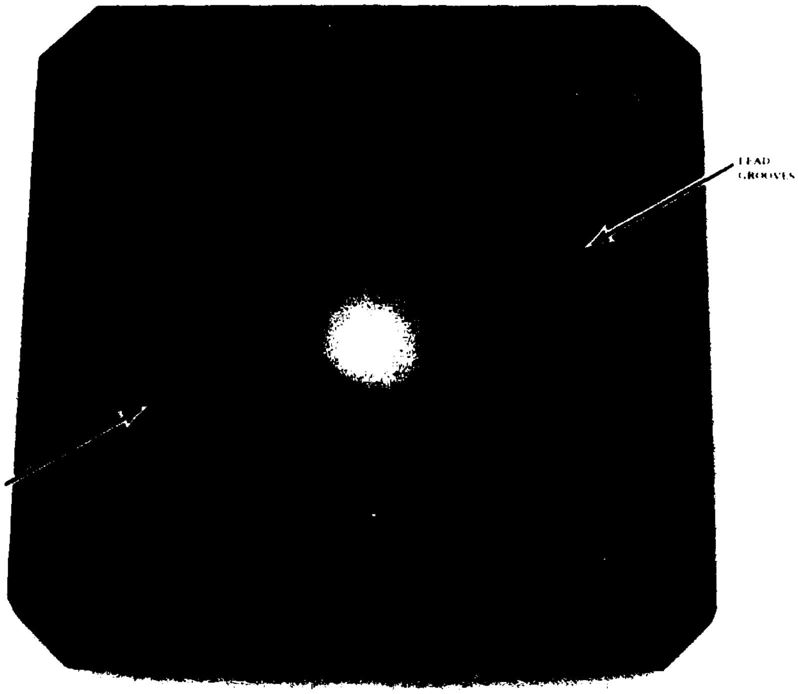


Figure 33. Side plate with 5-degree V-groove filed in the leading edge of the port to accommodate ease of start up.

Stop-start tests were conducted using the side plates with different port sweep angles. In the tests the test bench was set to different flow rates and then the flow directed to the motor. In all cases without the lead grooves, a 10% failure to start up the motor was experienced. That is, in 50 attempts to start the motor by applying seawater at 6 gpm it failed 5 times to start; instead, the motor locked up, and the system pressure rapidly rose until the relief valve operated. The addition of the 5-degree lead grooves reduced the failure to start to 2%.

A test was conducted with nonflexible side plates fabricated from Metcar 275. Details of the tests are described in the Motor Materials Evaluation section of this report. Briefly, no significant decrease was noted on motor performance over flexible side plates. Additionally, motor start-up torque was significantly reduced.

In the unidirectional motor, the spring holes were changed from the tapered shape shown in Figure 8 to a straight hole with the maximum diameter slightly less than that of the vane slot. The idea was to see if the taper reduced spring wear. The tests showed that spring wear was significantly greater with the straight hole. It was estimated that spring life was reduced from 100 hours to 50 hours.

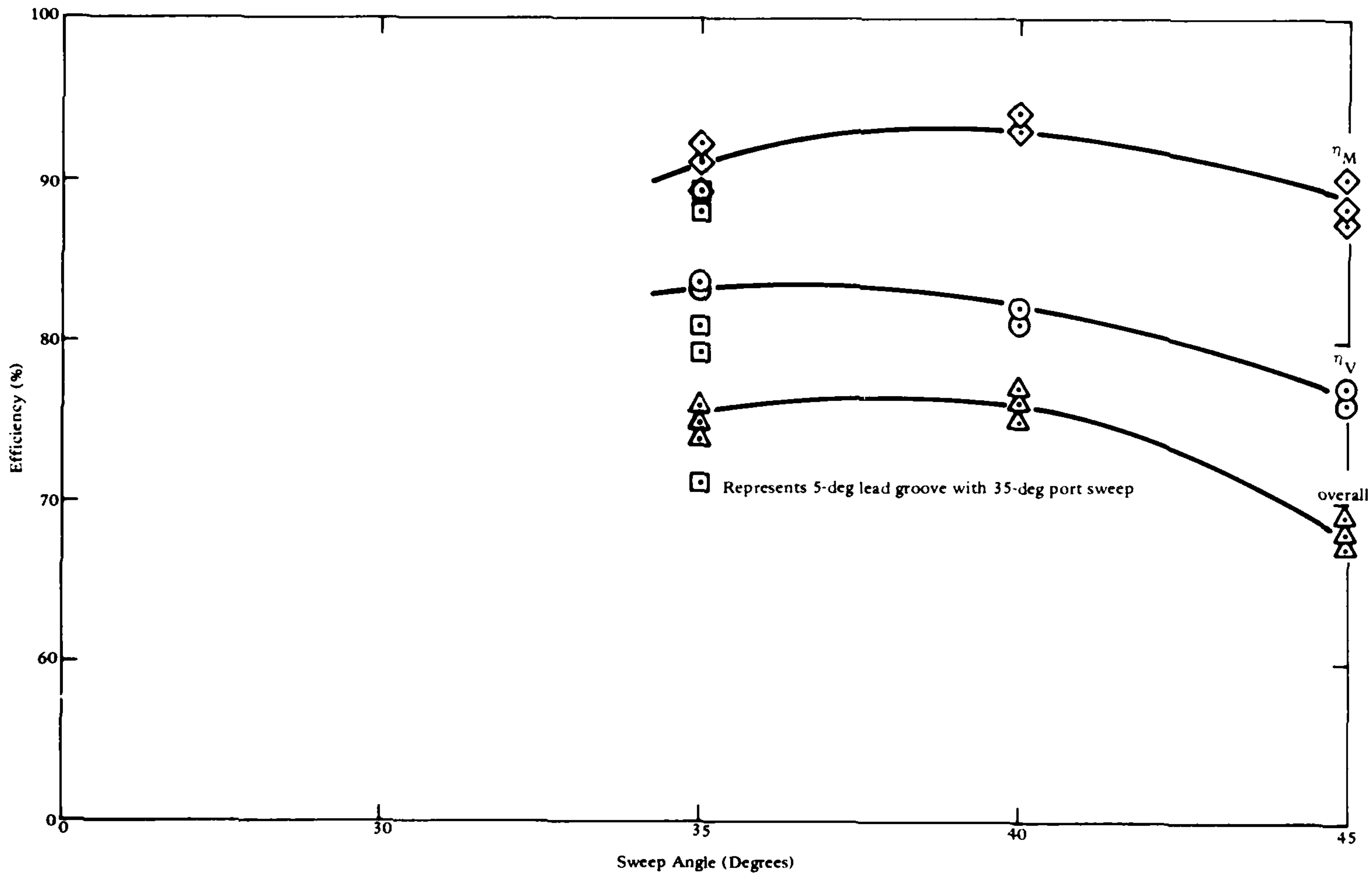


Figure 34. Performance of the reversible motor as related to port sweep angle.

Impact Wrench Tests

Motor-Impactor. The objective of the motor-impactor test was to obtain baseline data on the performance characteristics of the IR Model 2910 impact mechanism driven by the seawater vane motor. The primary concern was whether or not the motor could function adequately under the start-stop conditions imposed by the impact mechanism. The motor comes to a complete stop at impact once each revolution.

The arrangement for the motor-impactor tests is shown in Figure 35. A splined adapter shaft was used to connect the motor to the impact mechanism. Before the start of each test, a flow rate was set on the test bench. Test bolt tension and system flow rate and pressure were continually recorded using an oscillograph. An example of the recorded data is given in Figure 36. Since it was known that the mechanism would impact once each revolution, motor speed was determined by counting the number of impacts per second from the test data. Tests were run with the impact mechanism dry and flooded with seawater.

Initially the tests were run with a 3/4-12 test bolt mounted on the torque tester. During the first tests it was discovered that the motor-impactor output exceeded the strength of the test bolt, and the bolt was sheared off. The remaining motor-impactor tests were performed with a 1-12 bolt mounted in the torque tester. To eliminate further bolt failures, tests were terminated when the torque reached 1,200 ft-lb.

Figure 35. Physical arrangement of motor coupled to the impactor.

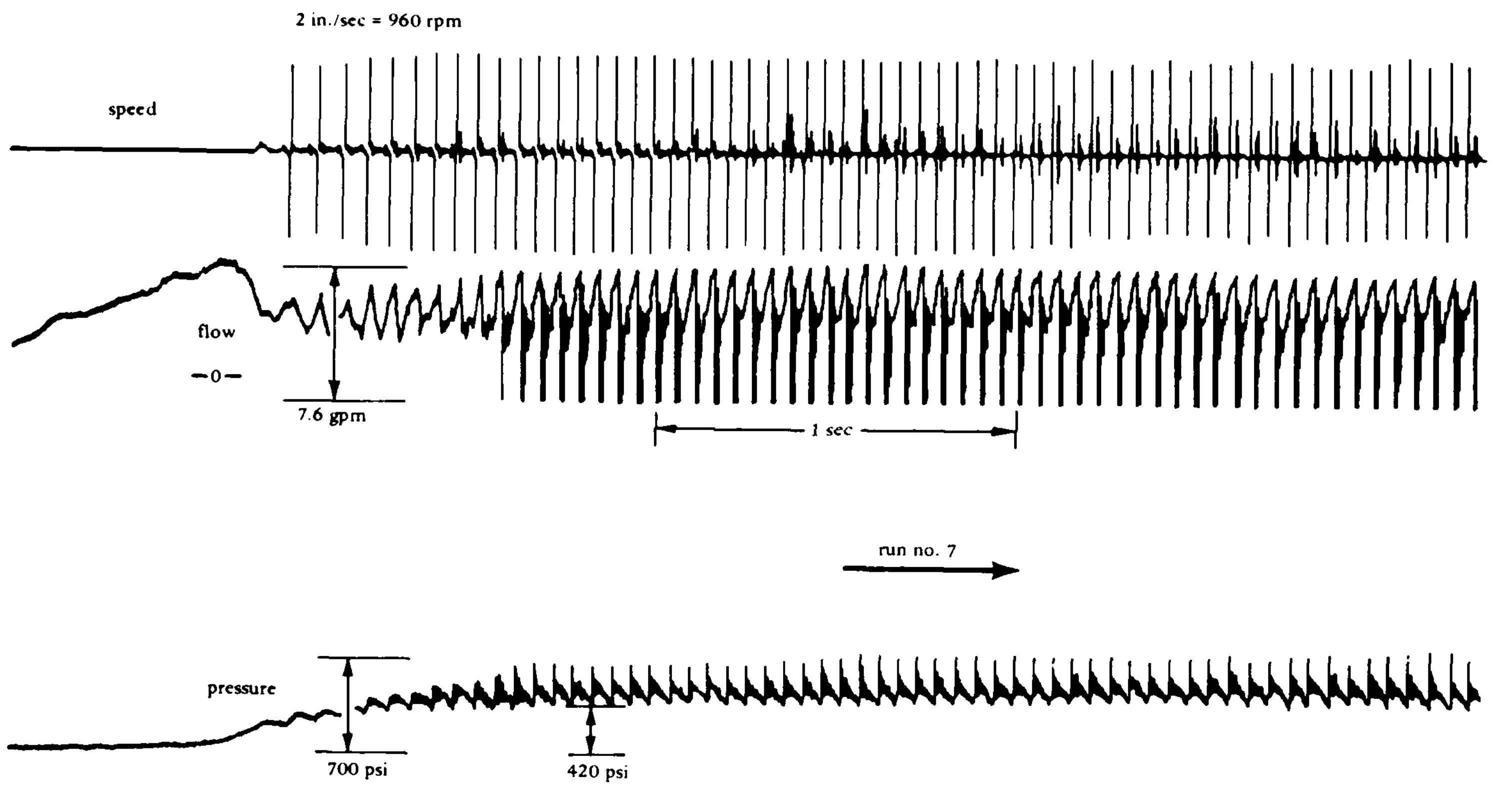


Figure 36. Data from impact tests. Note that flow, and thus RPM, goes to zero on every revolution.

Data from the tests are plotted in Figure 37. Overall, the tests proved that the motor-impactor combination is a very effective means of applying torque. The results were much better than initially expected. In fact, because of the manufacturer's recommendations, the motor-impactor test was limited to 1,200 ft-lb to protect the impact mechanism. Higher values could have easily been attained. In future tests an impact mechanism with greater output capacity will be used.

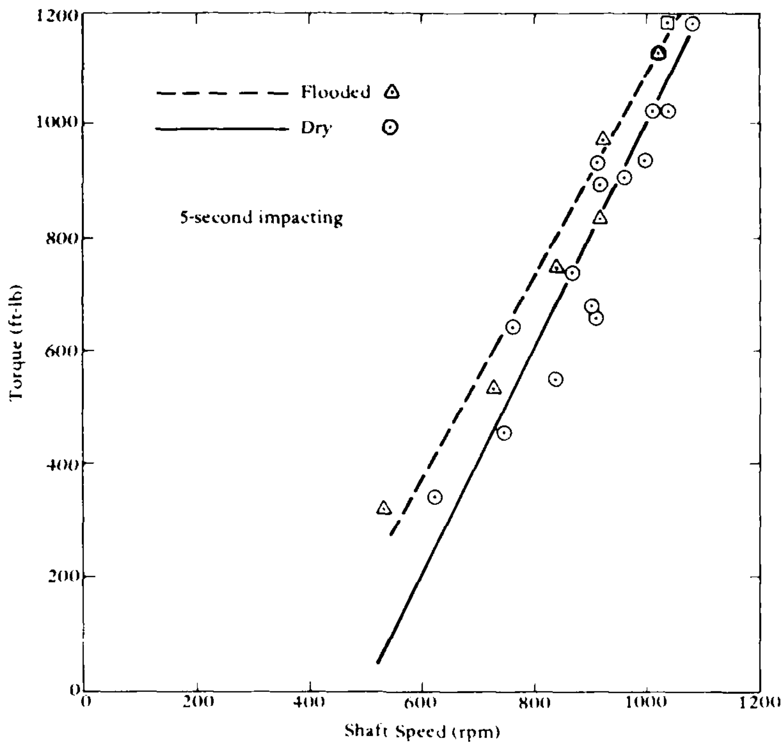
Output of the IR Model 2910 impact mechanism is dependent on rotary speed of the motor, which is determined by system flow rate. The time to reach a specific torque decreases with increasing flow rate. As seen in Figure 37a, little difference was noted in operation of the impact mechanism, whether flooded or dry.

Figure 38 shows a comparison of the output of the IR Model 2910 impact mechanism driven by the seawater vane motor and by a commercially available oil hydraulic gear motor. The seawater-impactor combination achieved higher torque for the same impacting revolutions per minute than the oil-impactor combination. These results are attributed to the fact that (1) the seawater motor rotational mass is much larger than that of the oil hydraulic gear motor and (2) the rotational mass of the motor is coupled to the impact mechanism during impact.

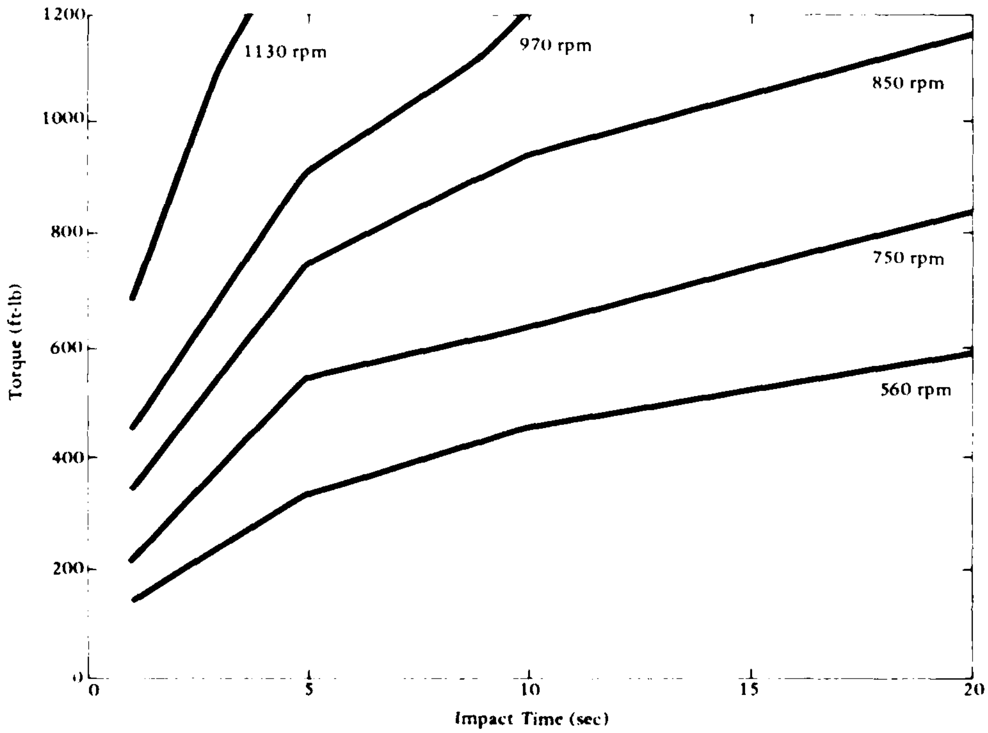
Motor Impact-Handle. Tests were conducted with the motor attached to the tool handle and mounted on the test bench. The objective of the tests was to obtain data on the performance degradation due to leakage and viscous losses from the control valves and porting of the handle. A comparison of the motor performance with the motor-handle performance is presented in Figures 39 and 40. At 6-gpm, 1,000-psi operation, volumetric efficiency of the motor-handle combination was 22% lower than motor efficiency, while mechanical efficiency was 8% lower. Volumetric losses were due to cross-port leakage in both the on-off and reversing spools. Mechanical efficiency losses were due to viscous line losses from internal passages within the handle.

Both mechanical and volumetric efficiency losses are greater than would be expected if the tool handle had been designed specifically for operation with seawater. The high volumetric losses were attributed to the short leakage paths in both spool valves. Also, prior to conducting the motor-handle tests the handle had been operated for approximately 10 hours in a test to determine the effect of flow erosion on the aluminum spool bore. Considerable pitting was found on the lands of the spool bores after this earlier test; this pitting effectively increased the width of the leakage path. The porting adapter plates that direct the flow from the handle to the dual inlet and outlet ports contributed to the mechanical efficiency losses by increasing flow restriction.

Assembled Impact Wrench. It was originally planned to conduct tests of the assembled wrench on the test bench, as well as in the water with divers. However, because of the low volumetric efficiency of the motor-handle assembly due to the poor condition of the spool valves, it was decided that the bench tests would serve no useful function.



(a) Impact torque achieved after 5 seconds of impacting for both dry and submerged impactor.



(b) Relationship between shaft speed and impact torque.

Figure 37. Data from motor-impactor tests.

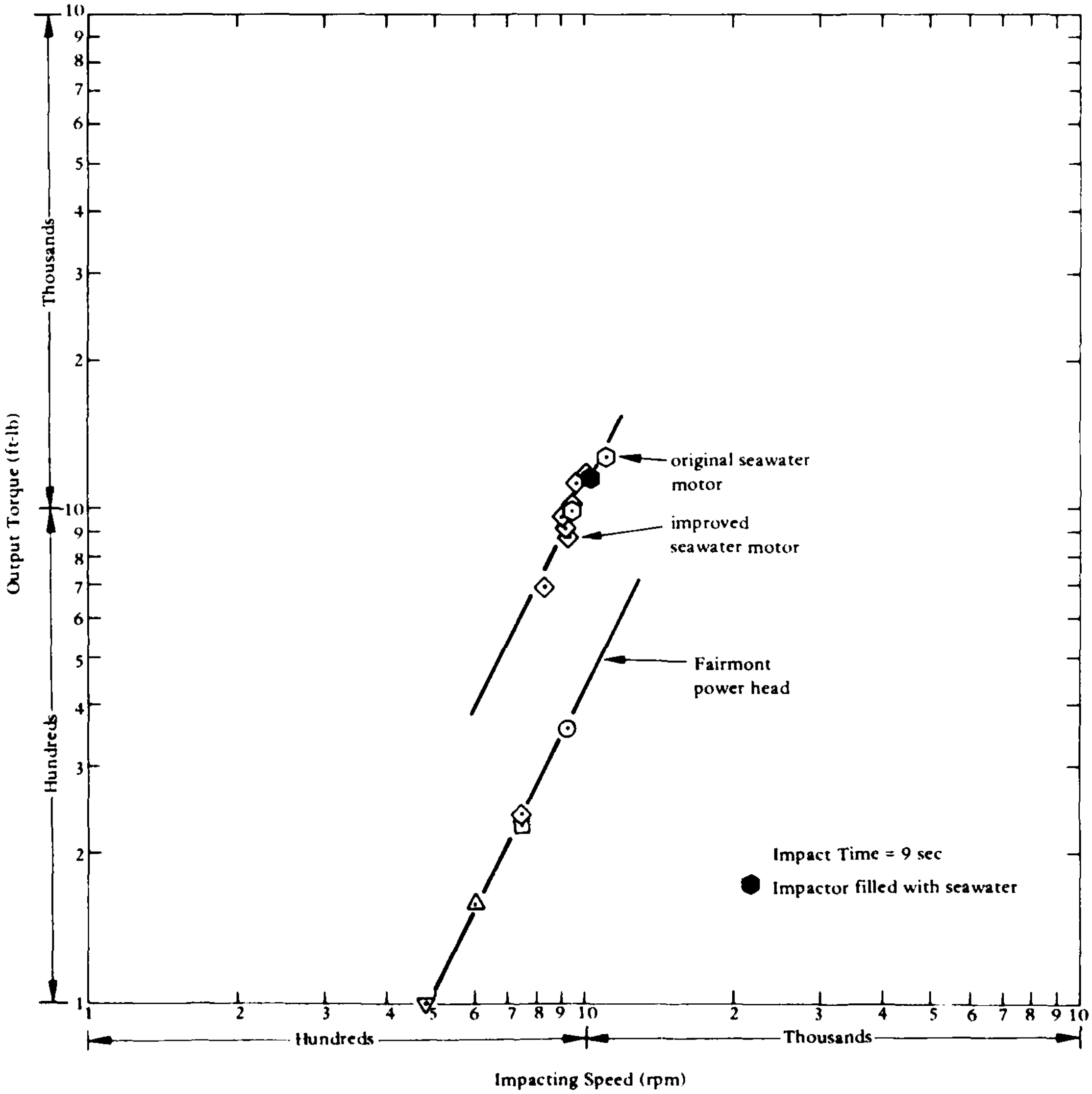


Figure 38. Comparison of IR Model 2910 impactor with different motors.

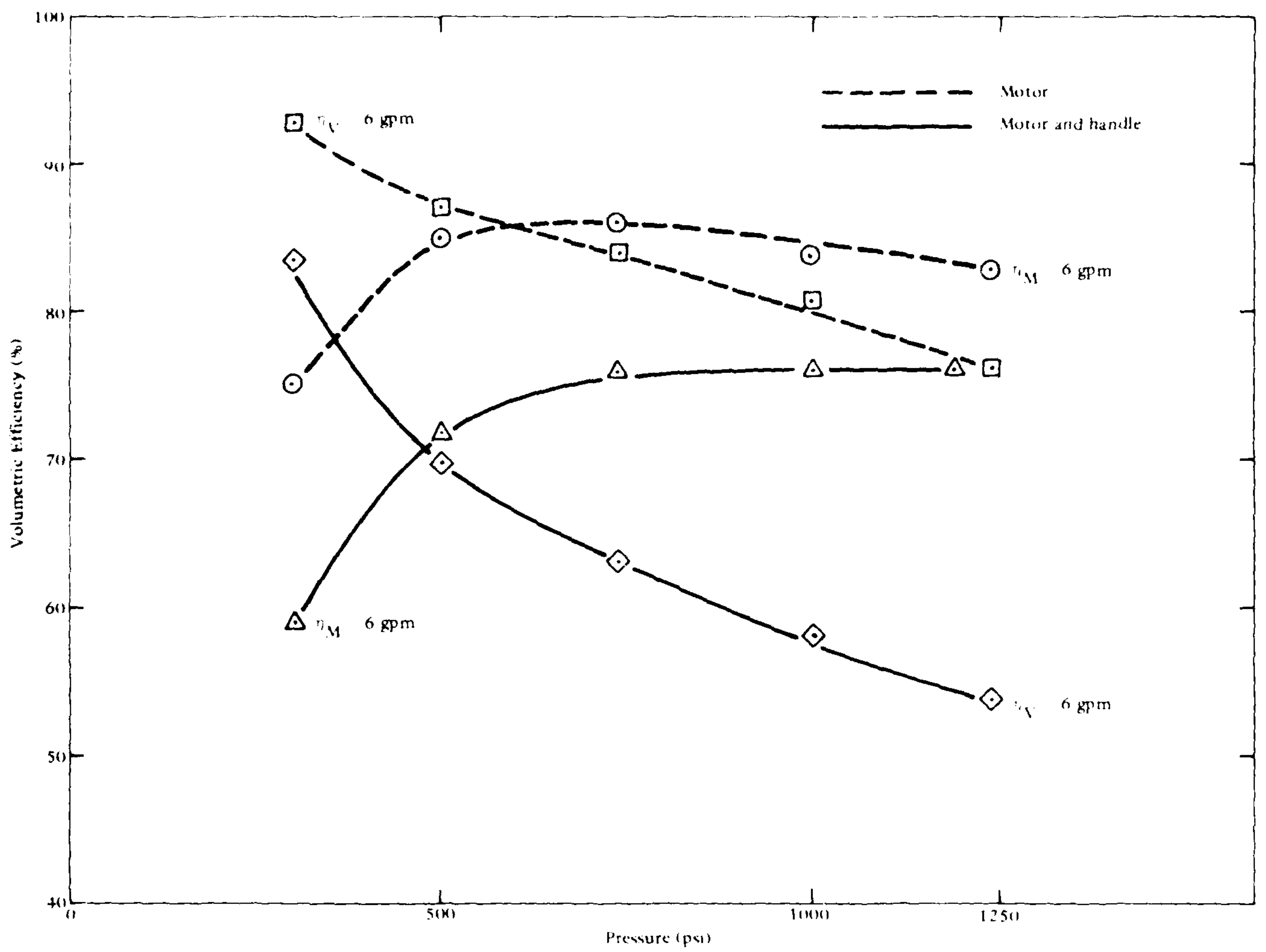


Figure 39. Volumetric and mechanical efficiency decreased significantly with motor installed on handle.

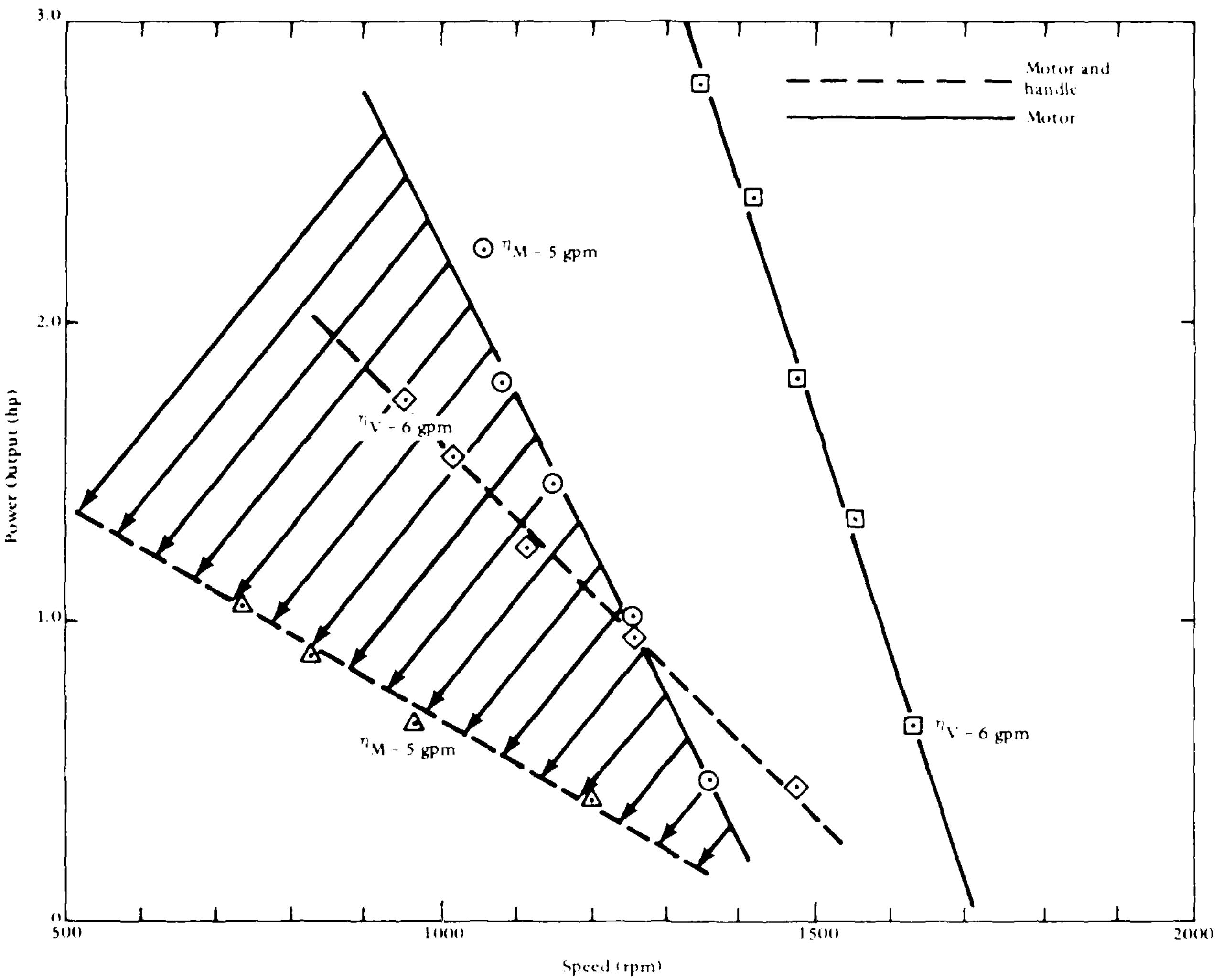


Figure 40. Comparison of power output versus speed for motor and motor mounted on handle

When the wrench was first assembled and tested, several problems were encountered. First, there was considerable leakage from the on-off spool valve. Second, when the wrench was operated, it ran irregularly and with low power; third, the motor would not always start when the valve was actuated. Investigation revealed that the motor was not adequately aligned with the impact mechanism. As a result, the intermediate shaft was binding in the adapter plate radial/thrust bearing. The misalignment caused excessive radial and axial loads to be transferred to the motor which caused the rotor to bind in the bearings. The alignment was corrected by placing dowel pins between the motor and handle.

Correction of the alignment problem solved the irregular running of the motor but did not affect the motor's starting problem. When the on-off valve was operated, system line pressure would increase to the relief valve setting, but the motor would not turn even under a no-load condition. It was thought that the problem might be due to a dead spot in the motor, wherein both sides of the power vane would become pressurized before rotation would occur. To eliminate this problem, small V-shaped grooves were filed in the motor side plates at each entry port. These grooves provided a low flow path 5 degrees ahead of each entry port. This modification reduced the motor start failures to less than 1 in approximately 20 starts.

After the problems were corrected on the assembled wrench, tests were conducted by Navy divers in the NCEL shallow water test tank. The wrench was powered by the SWPHS. The purpose of the underwater tests was to gain some operational experience using seawater hydraulic tools. Divers were asked to use the wrench to drill 3/8-inch-diam holes in mild steel plate, tighten and loosen nuts and bolts, and tighten the nut on the torque testing machine, which had been modified for underwater use. The underwater test setup is shown in Figure 41. To accommodate drilling, an adapter was attached to a standard key-type chuck shown in Figure 42.

When the wrench was first tested by divers, the motor worked properly but the wrench would stall when minor loads were applied to the output shaft. An investigation revealed that the impact mechanism was binding and causing the motor to stall. The impact mechanism was replaced with a spare, and the tests were successfully resumed. With 7-gpm input flow to the wrench, the diver applied 1,100 ft-lb of torque to the 1-inch-diam bolt on the impact test mechanism in 8 seconds and drilled 3/8-inch-diam holes in 1/2-inch-thick mild steel plate in 1.1 minutes.

The divers who tested the wrench were asked to comment on its handling. Despite the fact that the wrench was a prototype model and not human-factor designed, the divers had few problems in handling the wrench. The main comment was that the wrench was not balanced and was heavy on the motor end. The balance problem caused their wrists to tire after about 30 minutes of use. The divers commented that the exhaust at the tool handle did not cause handling problems; in fact, the divers mentioned that the warm water leaving the wrench might be beneficial in warming their hands in cold water.

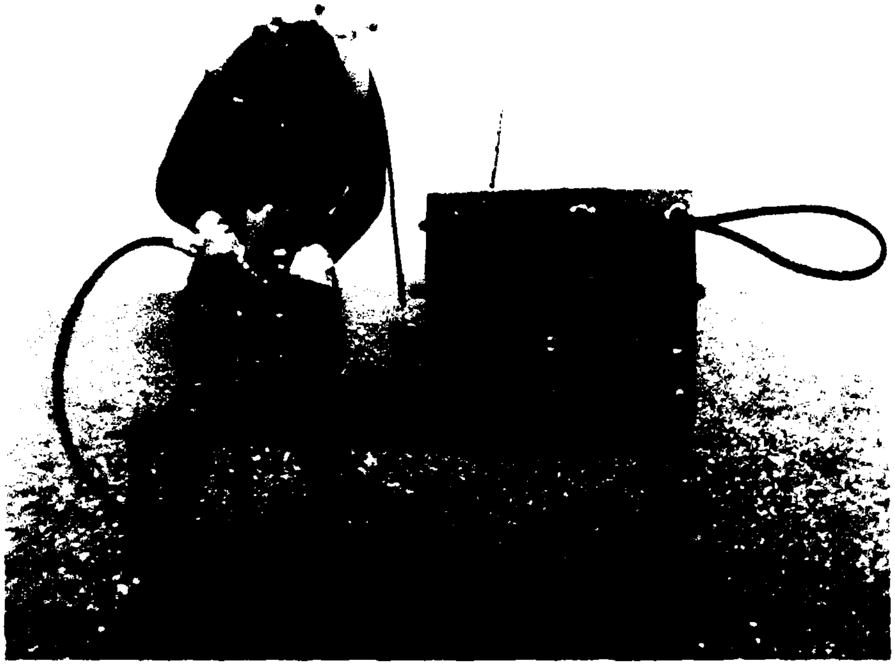


Figure 41. Underwater test setup for impact and drilling tests.

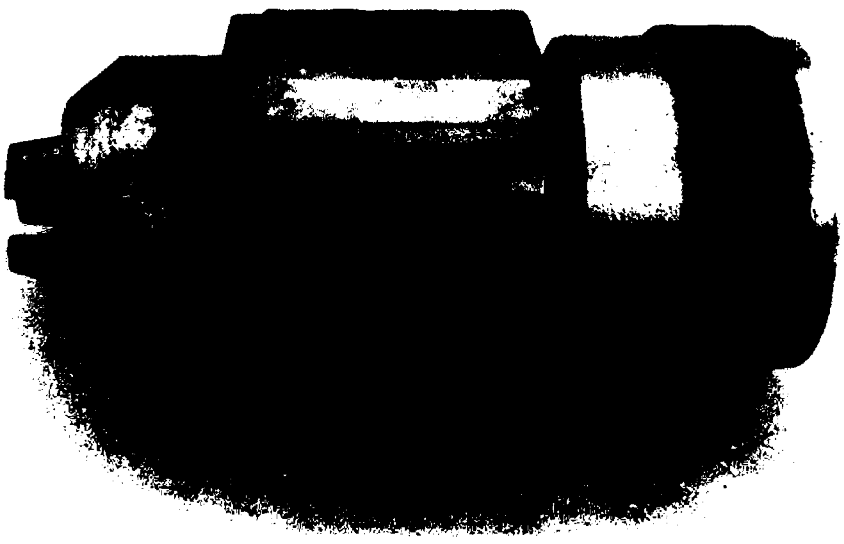


Figure 42. Adapter attached to key-type chuck used in drilling operation.

Propeller Cleaning Brush Tests

Laboratory Component Tests. Several different laboratory tests were conducted during the development of the cleaning brush. As with the impact wrench, all laboratory tests were conducted at the seawater test bench with components operated in air but flooded with seawater as necessary.

The first test was designed to determine the performance of the unidirectional seawater motor. The output from a typical test series is shown in Figure 43. Seawater flow rate was varied from 5 to 9 gpm, and pressure was varied from 250 to 1,250 psi. As can be seen in Figure 43, power output of the motor varied from 0.5 to 4.6 hp, and other power levels can be easily interpolated for any combination for flow and pressure.

The second series of tests conducted were with the motor and brush handle as shown in Figure 44. In these tests the handle had the same configuration as the impact wrench; that is, the forward-reverse valve was not modified. Seawater flow to the motor was controlled by the handle trigger valve. Results from this configuration are shown in Figure 45. It is quite evident that the motor power output was substantially lower than when the motor was operated alone For example, power output at 7 gpm and 1,250 psi was only 2.3 hp versus 3.6 hp without the handle - a 36% loss in power. These results prompted modification of the handle to reduce leakage as described previously.

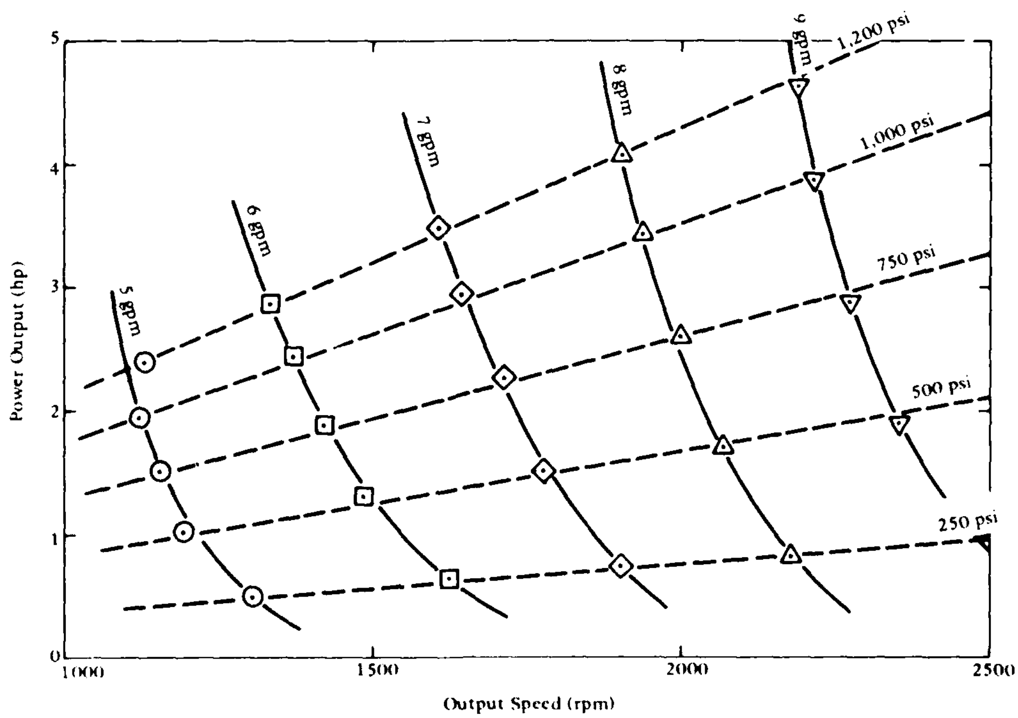


Figure 43. Unidirectional motor for propeller cleaning brush 35-degree side plates (second data set).

Figure 44. Motor and brush handle test setup

Performance tests were conducted with the handle modified to include the fixed and sealed reversing spool. The results are shown in Figure 46. This figure shows the power delivered at the brush's output shaft when operated in air. The brush operated satisfactorily over the range of conditions plotted. Also, the power reduction due to friction and flow losses in the handle were limited to less than 9% of the motor output power.

In-water Testing. Following the successful completion of the laboratory tests, the brush was tested in water. The brush was powered by the SWHPS through a single 1/2-inch-diam, 250-foot long flexible hose.

The purpose of these tests was to verify proper operation of the brush underwater and determine the power required to rotate the cleaning disk. Instrumentation was limited to pressure and flow rate delivered by the SWHPS and the brush output speed. The latter was measured by a stroboscopic tachometer aimed through a window in the dive tank. The pressure delivered to the brush was calculated by subtracting the pressure drop through the delivery hose from the pressure delivered at the

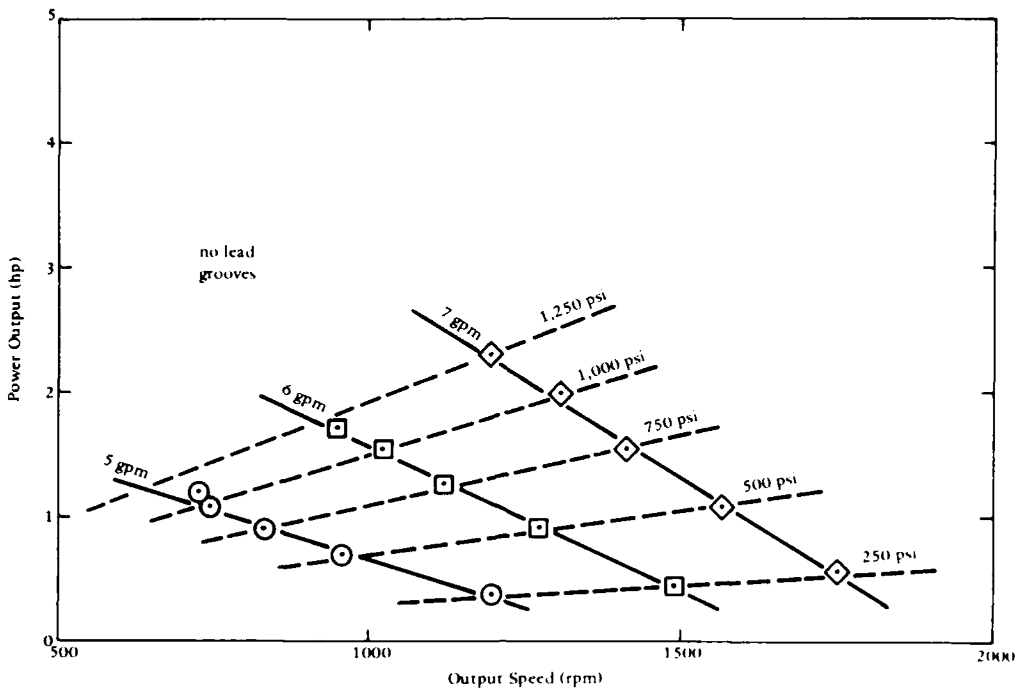


Figure 45. Seawater motor with handle test results. In this test the reverse spool was not sealed.

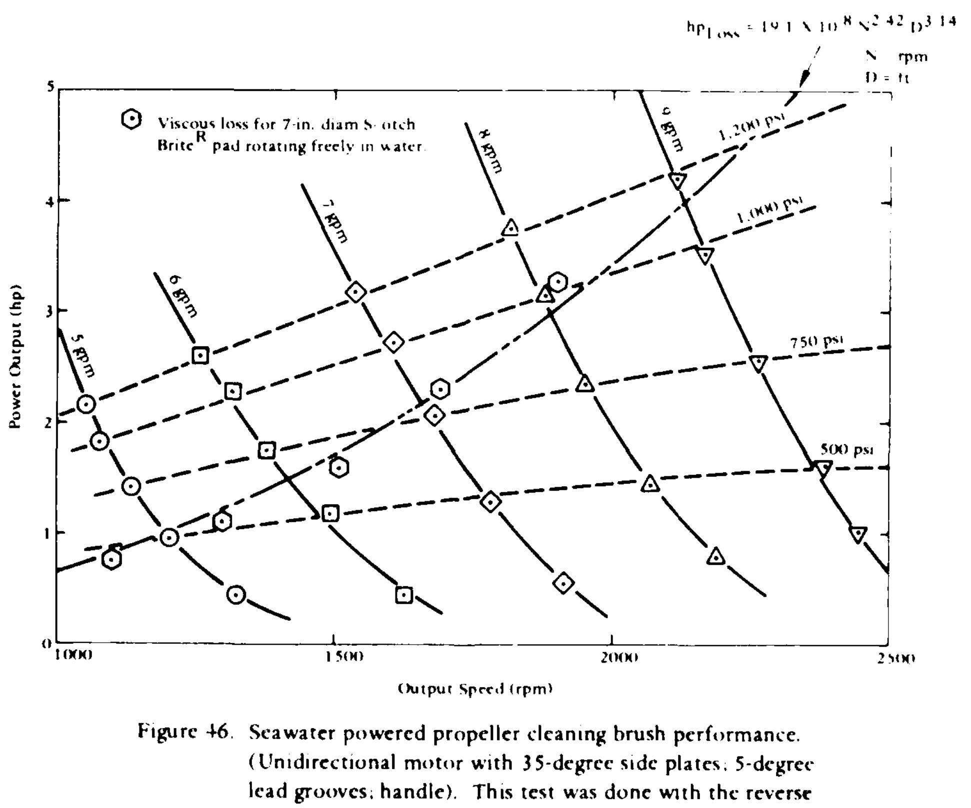


Figure 46. Seawater powered propeller cleaning brush performance. (Unidirectional motor with 35-degree side plates; 5-degree lead grooves; handle). This test was done with the reverse spool sealed.

SWHPS. Power consumed by the brush was determined by using the brush's characteristic curves, shown in Figure 47, where the flow rate, pressure drop, and output speed are known and the power output can thus be read directly.

The measured relationship between flow rate and delivery hose pressure drop is shown in Figure 48. These results are independent of the brush employed and are applicable to all future tests with this delivery hose.

The cleaning disk selected for use was a 7-inch-diam Scotch Brite[R] pad mounted on a standard rubber disk (similar to those used with disk sanders). The Scotch Brite pad-disk was selected because it would require less power to rotate underwater than a wire brush; a particular concern since the motor would require substantial redesign to increase output power significantly. The brush was first operated freely in air with and without the disk attached. The pressure drops were compared for these two cases, and it was found that there was no measurable difference; i.e., the power required to rotate the disk in air was negligible. Next, the brush was operated freely underwater without the disk attached, and a slight pressure drop was noted. Accurate estimation of power at this low level was difficult due to the measurement technique employed, but suffice it to say that the power increase was less than 0.2 hp.

The disk was then attached, and the brush again operated freely underwater. As was expected, the brush's pressure drop and the power required to rotate the disk in water was substantial. The relationship between this viscous power loss and the disk speed is shown in Figure 47. The loss is very sensitive to the disk speed, and the exponent of 2.42 is identical to the value that can be obtained from data on abrasive wheels contained in Reference 4. Data in Reference 4 also show that the viscous power loss is related to the abrasive wheel diameter by the 3.14 power.

Data obtained for the viscous loss from the Scotch Brite pad are plotted in Figure 46. Also shown is a best fit curve using the above exponents. Careful evaluation of this figure reveals much about the ability of the brush to perform useful work underwater. For a flow rate of 5 gpm and a maximum allowable pressure drop of 1,200 psi, the brush equipped with this disk will produce a maximum usable power of 1.2 hp (2.2 - 1.0 = 1.2 hp); but at 9 gpm and the same pressure, the maximum usable power is only 0.25 hp (4.25 - 4.00 = 0.25 hp). Hence, even though the cleaning disk may be more efficient at higher speeds, the overall cleaning ability would be less because much less net power is available to the brush. In short, more power, higher speeds, and bigger disks do not make a better scrubber.

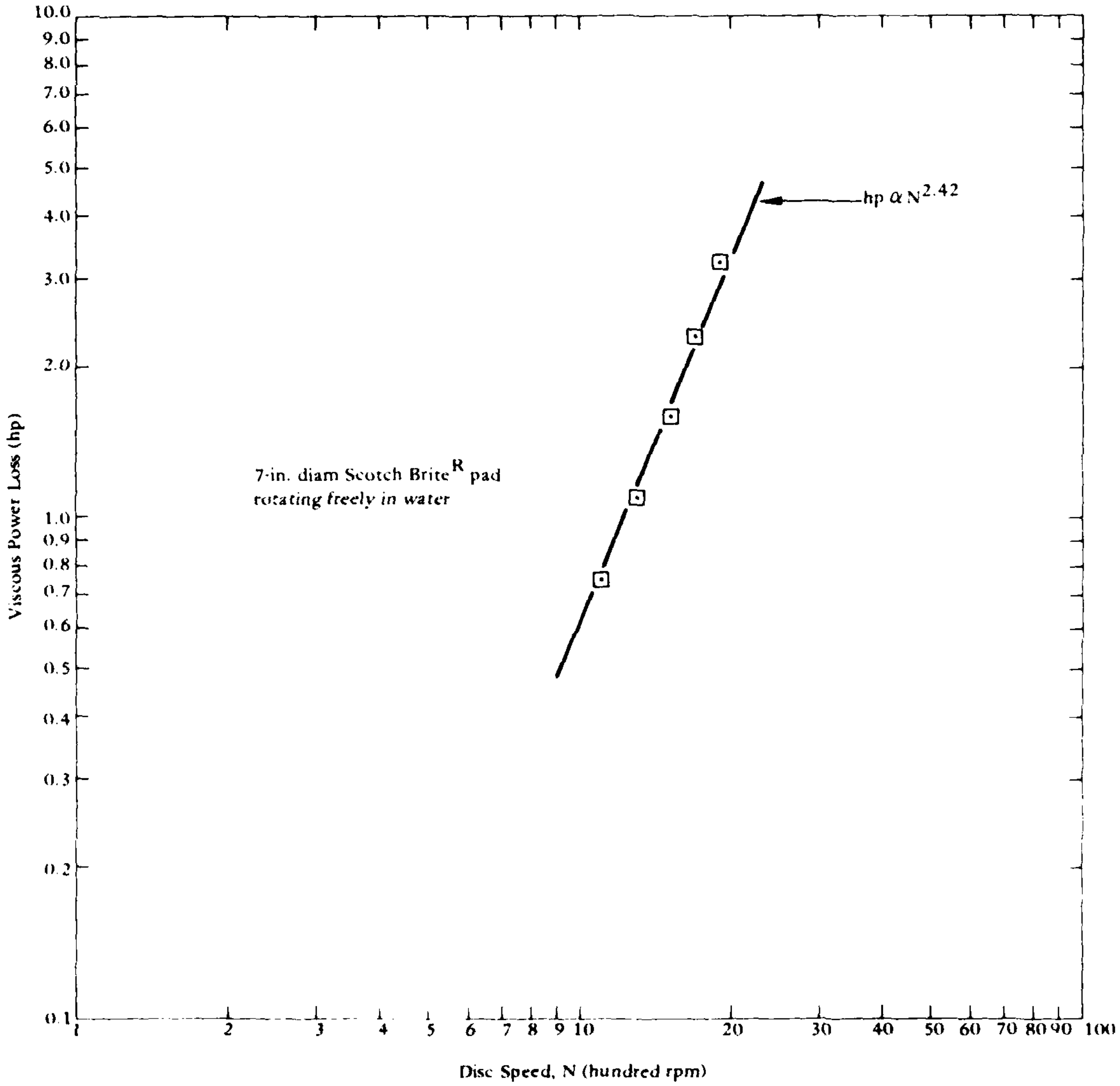


Figure 47. Cleaning brush's characteristic curve of power required to rotate at a given speed in water.

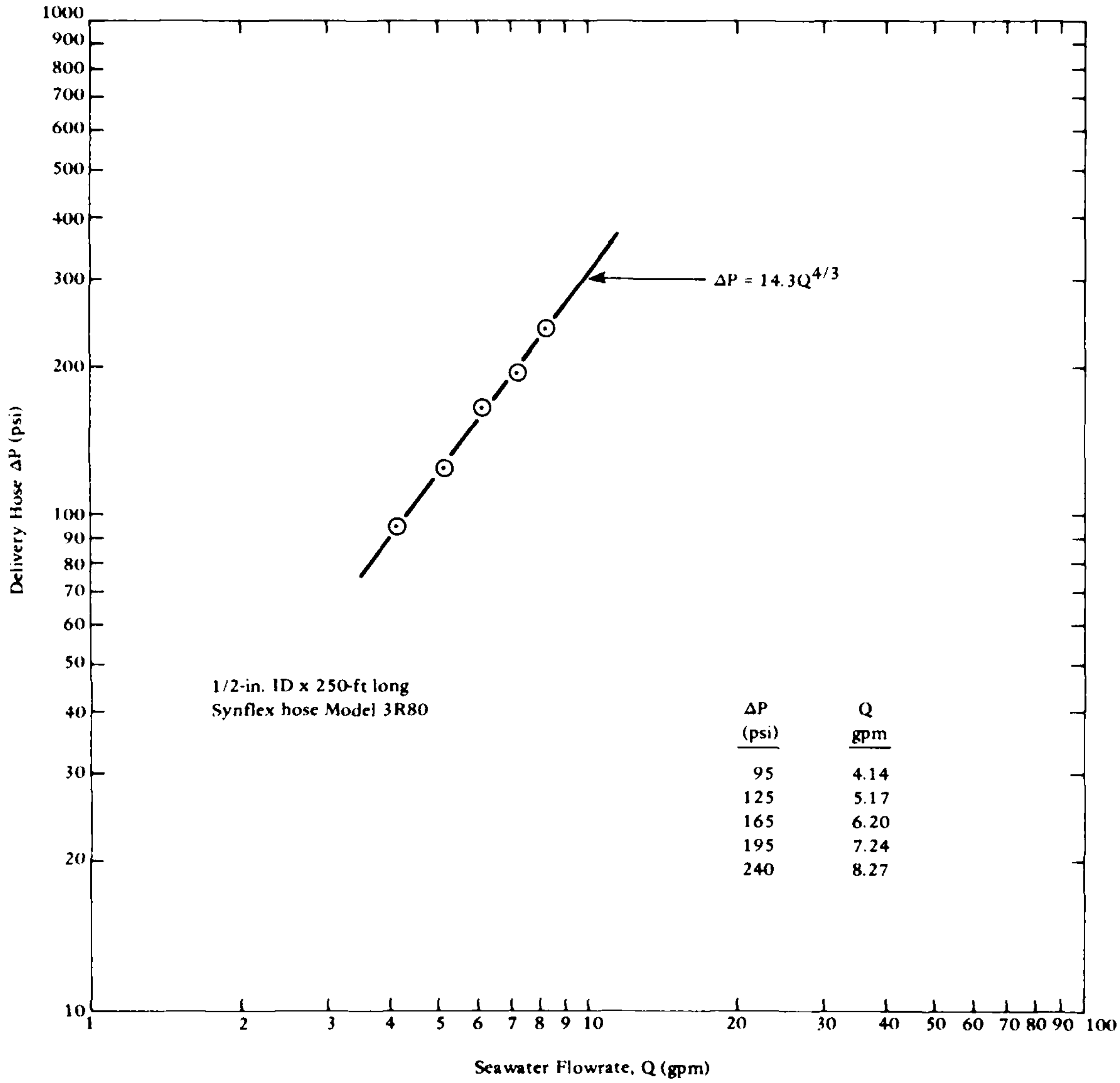


Figure 48. Measured relationship between flow rate and delivery hose pressure drop.

To verify this finding, five different divers performed an informal evaluation of the brush by cleaning sections of the dive tank wall (see Figure 49). Their unanimous conclusion was that the brush did not clean any better at higher flow rates than at lower ones. At higher flow rates the brush was much more difficult to control, probably due to the higher viscous disk losses which the diver had to counteract. Other comments by the divers were:

- They are generally pleased with the brush.
- The easiest way to control the brush and counteract the torque is with the assist handle opposite the pistol-grip handle.
- The brush is too heavy to operate above the heart for very long but can be operated below the waist without difficulty.
- The noise is quite acceptable.
- The finger trigger is somewhat tiring; a palm trigger would be better.
- The brush tends to wobble when moving to the left; a more flexible backing pad might help.
- The delivery hose is stiff, making it awkward to operate with the pistol-grip handle in the horizontal plane; a hose swivel at the handle would be nice.

Seawater Hydraulic Power Supply Tests

Tests were conducted on the SWHPS to determine output, performance of system components, and operability and to identify problem areas that would require development.

The objectives of the first test were to determine the maximum output of the SWHPS and performance of the pressure-compensated-flow-control valve. Not all diver tools will require the same flow rate for proper operation. However, each specific tool will have a particular flow rate requirement. To accommodate this requirement, the SWHPS was equipped with a flow controller, which automatically maintained system flow rate regardless of the operational pressure. For normal operations, it is satisfactory if the flow controller maintains the desired flow within 0.5 gpm.

The flow rate controller and the system power tests were conducted with the SWHPS connected to the load circuit as shown in Figure 50. The circuit contained a calibrated flow meter for verification of system flow rates. The SWHPS flow controller was set to a specific flow rate at 500 psi, and pressure was changed by adjusting the load simulating needle valve. Data on pressure and flow rate were visually observed and manually recorded.

Figure 49. NCEL diver using cleaning brush on the wall of dive tank.

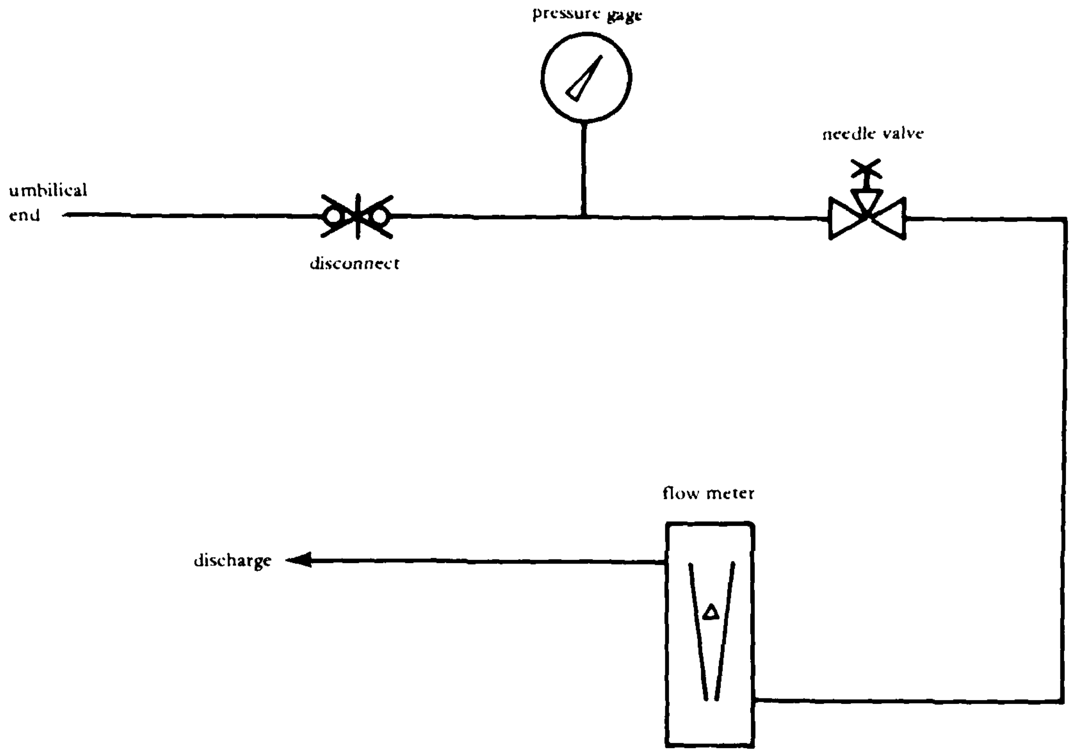


Figure 50. Test equipment schematic.

Performance of the SWHPS flow controller is shown in Figure 51. The results show that the flow rate was controllable within the requirements for diver tools. Maximum flow rate of the system was determined to be 12 gpm. Although the pump is capable of operating to 3,000 psi, tests were limited to 2,250 psi.

Several minor but annoying problems showed up during the SWHPS performance tests: first, the filter system continually leaked; second, some of the stainless steel tubing parted because of improper installation and possibly vibration from the diesel engine; and third, air had to be bled from cylinders of the main pump.

After the filter system was disassembled, it was revealed that the leakage was due to poor workmanship on the specially fabricated fiberglass housings. Furthermore, the filter elements selected were not compatible with the seawater environment but were constructed from zinc-plated mild steel cages with pleated-paper filter elements. The filter system was later replaced with a commercially available system (Figure 52), which consisted of a nylon mesh bag inside of a tubular stainless steel housing. With this system, the bags could be easily replaced, and filtration levels to remove particles down to a 5μ nominal size were readily available.

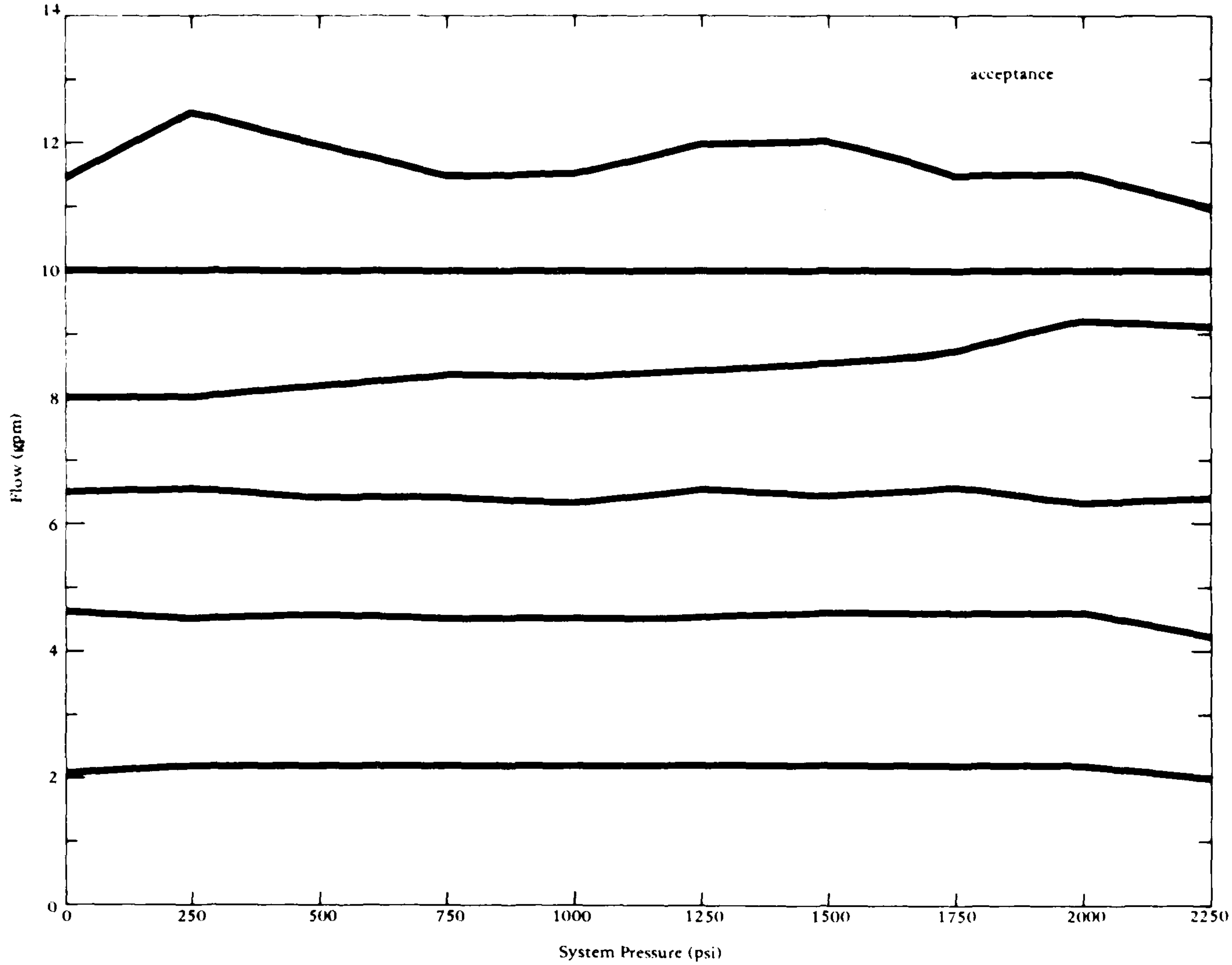


Figure 51. Performance results of the SWHPS flow controller.

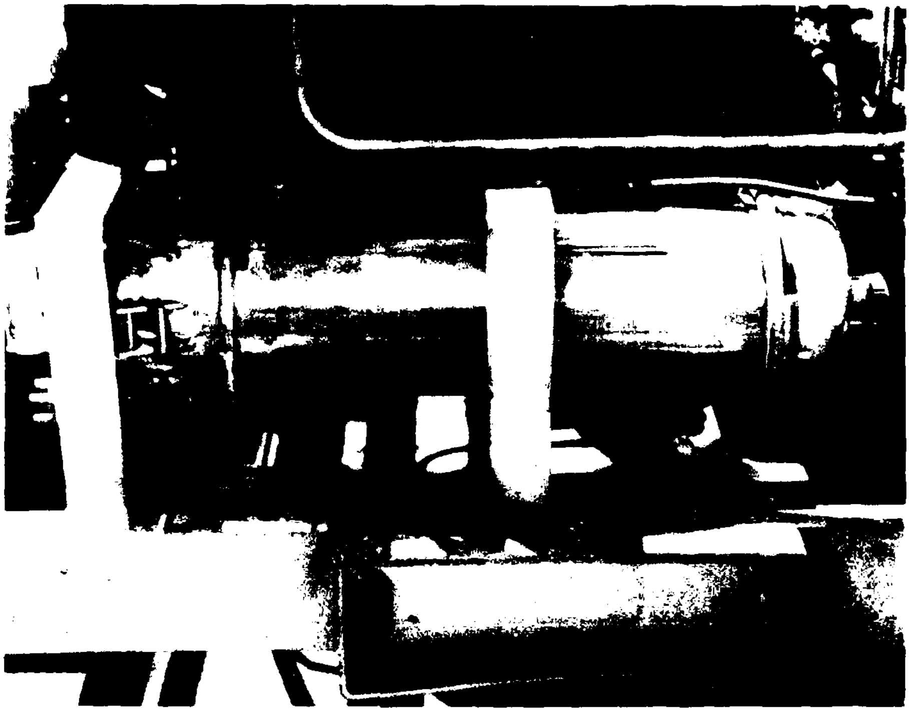


Figure 52. Tubular stainless steel filter system. Five micron mesh bag filter located inside canister.

The problem of air trapped in the water side of the pump pistons occurred frequently after the pump had been sitting idle overnight. The result of air trapped in the pump was a pulsation of the seawater output flow and sounded much like a water hammer. The pulsation was severe enough to damage the system's instrumentation. The hammering was eliminated by bleeding the air from each of the pistons while the system was running. This operation was quite awkward because of the location of the bleed screws and often took about 30 minutes to accomplish. The reason for air being trapped was not determined. With the pump properly bled its pump output was relatively pulsation free.

A test was conducted to determine the capability of the heat exchanger to maintain reservoir temperature under various operational conditions. If the water temperature at any time rises above 150°F, wear on the plastic vanes and bearings accelerates, and motor life is reduced.

The results of the reservoir temperature test are shown in a plot of reservoir temperatures for various flow conditions (Figure 53). The worst condition is when the tool is not used and all of the system's flow (12 gpm) is returned to the reservoir with an initial pressure of 1,500 psi. In this case, approximately 27,000 Btu/hr are being fed to the heat exchanger from the back pressure relief valve. The test showed that the reservoir temperature rose by 40 degrees. This rise should cause no problems in the tool's operation; in fact, the warm water may be beneficial in some conditions where the divers are submerged for extended periods of time. For example, the diver could place his hands in the exhaust stream to warm them.

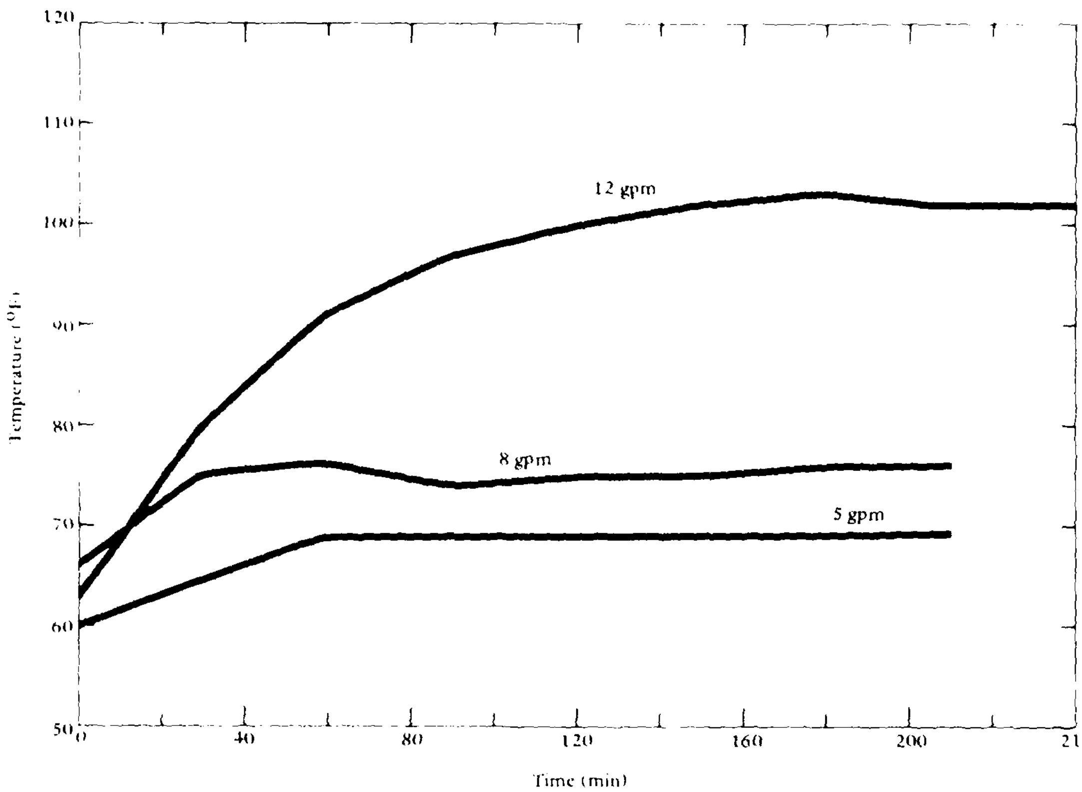
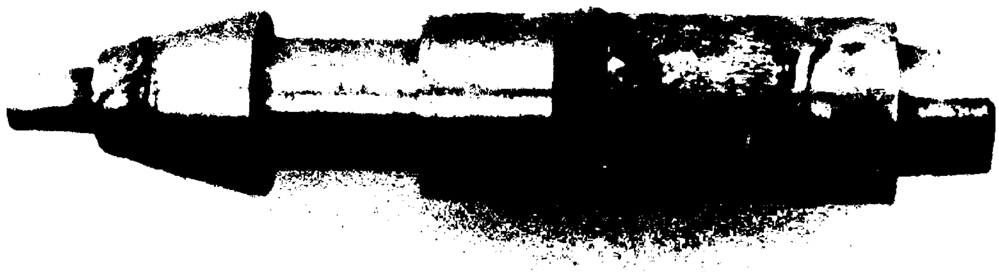


Figure 53. Reservoir temperature test results. Flow indicated is amount that is returned to the reservoir.

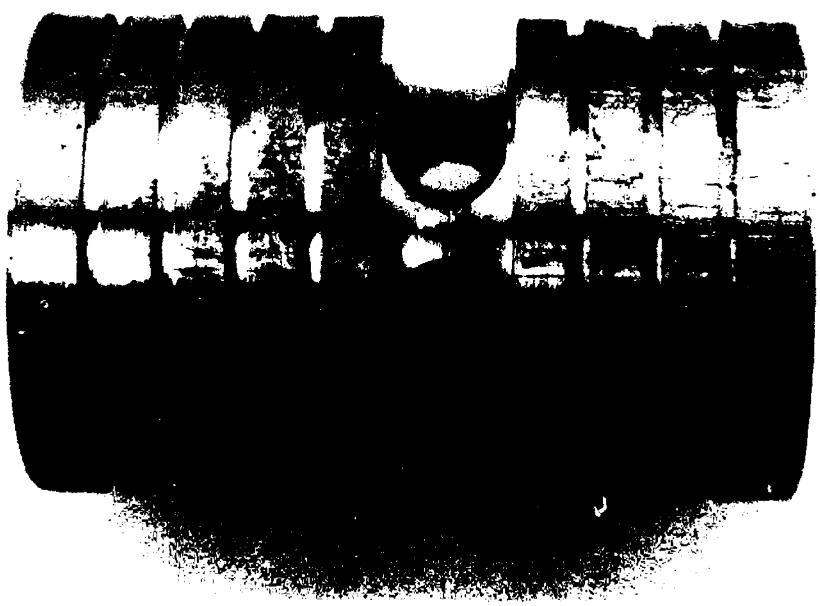
Both the back pressure regulator and the flow control valve failed after 50 to 60 hours of operation. The valves had been designed especially for the SWHPS application. The design basically involved modifying the oil hydraulic design with materials substitutions, and reduction in clearances between internal components. After the back pressure regulator and flow control valve were disassembled, it was discovered that the piston and sleeve (Figure 54) were severely galled, and motion was restricted. Galling was due to the use of similar stainless steel materials (Type 316) on two components that did not have adequate clearance and continually rubbed together.

The back pressure regulator was cleaned, repaired, and reinstalled. The flow control was replaced with a commercially available unit (Figure 55). This particular flow control valve was chosen because there were no rubbing metal-to-metal components within the valve, making it relatively immune to contamination. After the initial failure of the back pressure regulator, the power source was run for an additional 70 hours without failure.

In all, the SWHPS was operated for 240 hours under loaded conditions. Aside from the problems described, the SWHPS functioned satisfactorily. No major problems were encountered, which suggests that a major developmental effort is not necessary.



(a) Piston from back pressure regulator was severe [illegible]



(b) Galled piston from flow control valve.

Figure 54 Damaged piston and sleeve after 50 to 60 hours of operation.

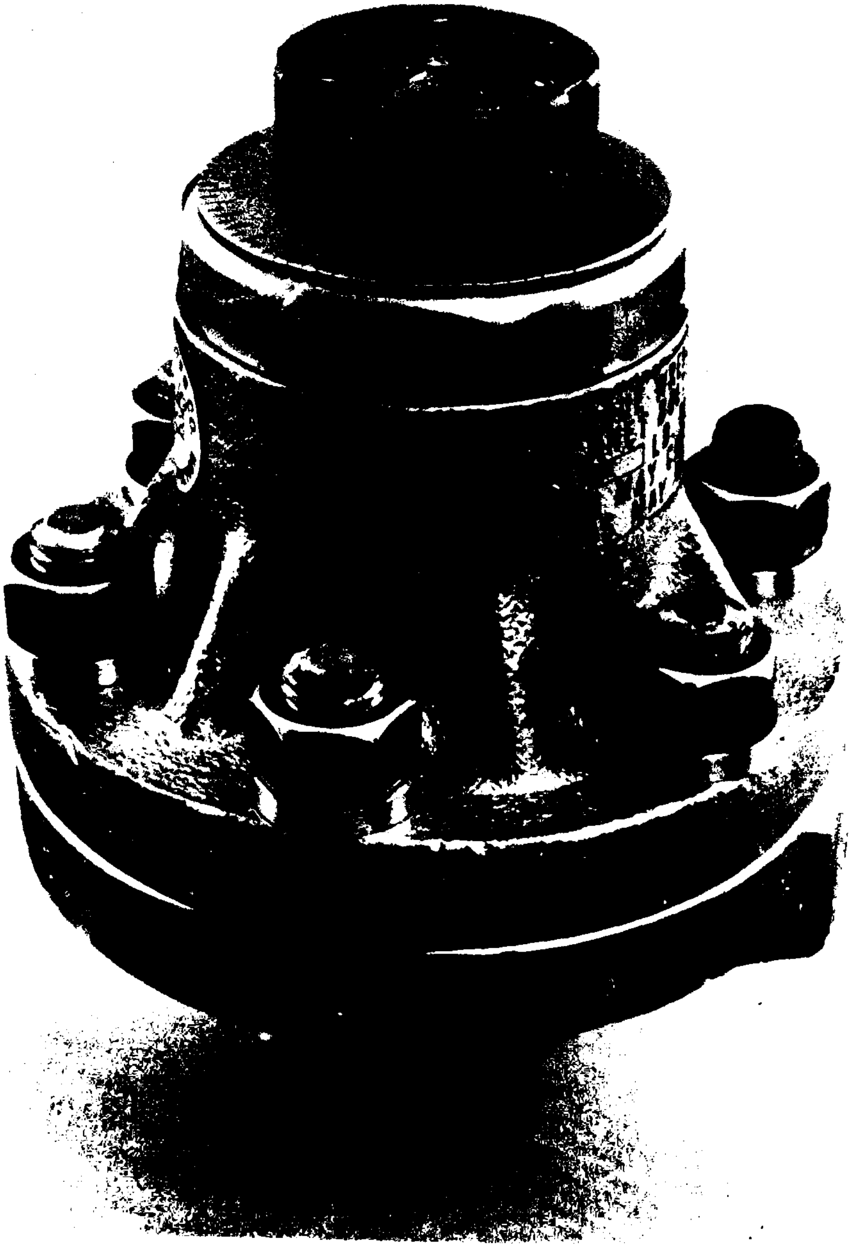


Figure 55. Kates flow control valve.

SUMMARY

The overall objective of the research described in this report was to demonstrate the feasibility of developing a diver tool system that uses seawater as the working fluid. Two improved motors for powering the rotary impact wrench and rotary propeller cleaning brush were developed and tested. A portable diesel-driven seawater hydraulic power supply was also developed and tested.

The reversible motor measures 3 by 3 by 3-1/2 inches and weighs 6.5 pounds. The unidirectional motor measures 3 by 3 by 3 inches and weighs 4-1/2 pounds. The additional weight of the reversible motor is due to the added components required to lubricate the bearings for the reversible operation.

Laboratory tests of the reversible motor included determining the operational characteristics and evaluating alternative materials for vanes and flexible side plates. The motor was also successfully evaluated in a 200-hour endurance test. Tests revealed that the reversible motor could provide the designed power of 3 hp with greater than 70% overall operational efficiency and could be operated for 100 hours before the springs failed. All other components of the motor operated for more than 200 hours without failure.

The reversible motor was coupled to a rotary impact mechanism and piston-grip handle to provide a breadboard underwater impact wrench, which was laboratory tested and diver evaluated. The impact wrench weighs 17 pounds in air and is 13 inches long. The seawater hydraulic impact wrench was not optimally designed for either diver use or long term submerged operation. Instead, to minimize development cost and time the pistol-grip handle, operational spool valves, and adapter plates were fabricated using components from off-the-shelf oil hydraulic tools. Tests involved operating the impact wrench mechanism against a 1-inch-diam bolt to determine tightening capabilities. With an input flow of 7 gpm the diver was able to apply 1,100 ft-lb of torque to the 1-inch bolt in less than 8 seconds. The diver was also able to drill holes in a mild steel plate with the impact wrench.

With the unidirectional motor for power, a rotary disk propeller cleaning brush was developed and tested. The brush weighs 12 pounds in air, 8-1/2 pounds submerged, and is 11 inches long. The brush uses the same pistol-grip handle as the impact wrench, except that the reversing spool valve was made inoperable and sealed to reduce high pressure leakage and improve efficiency. Underwater testing of the brush showed that considerable power is required to rotate the 7-inch-diam propeller cleaning disk in water. For example, the viscous losses in turning the disk at 1,800 rpm in water required approximately 2.7 hp. At 8-gpm flow rate and 1,200 psi, the brush delivered approximately 3.5 hp at 1,800 rpm. Comparison of these results shows that a net 0.8 hp was available at this speed for the diver to do useful work and that the motor was adequately powered but not overly powered for this type of work.

The SWHPS was developed to provide a portable means of delivering pressurized seawater to the tools. The SWHPS is diesel driven, weighs 3,500 pounds, and measures 192 inches long by 80 inches wide by 70 inches high. The SWHPS was not designed to be compact but rather for

easy accessibility to the components to facilitate laboratory evaluation. At full power the SWHPS can lift seawater from the surface to a height of 15 feet and deliver it to the 250-foot-long umbilical hose at pressure-compensated flow rates to 12 gpm and pressures to 2,000 psi. Filtration to 5μ nominal size is included. Tests of the SWHPS included evaluation of system components and overall performance. With minor modifications the SWHPS was capable of providing the required flow and pressure.

CONCLUSIONS

1. The feasibility of seawater hydraulic diver tool systems has been clearly demonstrated by the successful development and evaluation of an experimental portable, diesel-driven power source, rotary impact wrench, and propeller cleaning brush.

2. Operational characteristics of both the reversible and unidirectional balanced vane motors met or exceeded the developmental objectives of providing up to 3 hp with greater than 70% efficiency.

3. The experimental rotary impact wrench functioned adequately during the laboratory evaluations and provided sufficient torque to meet most future Navy diver requirements. The assembled wrench was capable of providing up to 1,100 ft-lb of torque in 8 seconds with 7-gpm seawater when used by divers. Higher torque could have been achieved but was limited by the manufacturer's recommendations for the impact mechanism.

4. Further development of the rotary impact wrench is needed to provide a pistol-grip handle with on-off and reversing controls suitable for diver use for long periods of time. Increasing the efficiency of the control valves will greatly improve the wrench's performance. It is expected that the overall size and weight of the wrench will be reduced in future models by eliminating the need for the porting adapter plate and impact mechanism adapter.

5. The propeller cleaning brush functioned exceptionally well during the limited evaluation. Modifying the pistol-grip handle to eliminate the leakage at the reversing spool valve greatly increased the brush's efficiency over that of the impact wrench. The divers were able to operate the tool underwater with only minimal problems. Exhausting the fluid from the motor's outlet ports did not cause problems for the divers. The unidirectional motor provided sufficient power to rotate the disk at speeds to 2,000 rpm while still providing at least 0.5 hp for conducting the cleaning operation. If the brush is used at greater speeds a larger motor will be required or a smaller diameter disk will be needed.

6. The SWHPS effectively provided sufficient power for operating the tools. The present configuration is adequate for laboratory test and evaluation. For field use by Navy divers, the SWHPS will have to be lighter and smaller. The present pump is capable of providing up to

12-gpm pulse-free seawater at 2,000 psi on a continuous basis. The problem of having to bleed air from the pump after it sits unused for prolonged periods of time will have to be solved before the SWHPS is satisfactory for Navy diver use. The pressure-compensated flow control and back-pressure regulators failed due to material and design problems. Modifications are necessary to provide suitable valves for future units. Overall, the flow circuit operated satisfactorily. Future designs should emphasize simplifying the circuit to reduce the number of components required.

7. Improvements in vane spring design increased the motor's life from 50 to 100 hours at full power operation. Other components of the motor have a service life expectancy of greater than 200 hours. Further improvements to reduce or eliminate the rubbing wear of the spring in the rotor will increase the overall service life beyond 200 hours. Based on the author's experience it is expected that a service life of 500 hours can be expected.

8. The small size and low weight of the experimental motors make them well-suited for the diver tools application.

9. The bearing lubrication system for the reversible motor is not satisfactory due to the corrosion and contamination sensitivity of the flow restrictors. A better system needs to be devised.

REFERENCES

1. Black, S.A., Seawater Hydraulics: Development of an experimental vane motor for powering diver-held tools, NCEL Technical Report R 895, July 1982.

2. Naval Civil Engineering Laboratory, Contract Report CR 80.015, Development of a seawater hydraulic vane motor for diver tools, Westinghouse Electric Corporation, Oceanics Division, April 1980.

3. Naval Civil Engineering Laboratory, Contract Report CR 82.025, Development of a seawater hydraulic tool system, Westinghouse Electric Corporation, Oceanics Division, May 1982.

4. Schlechting, H. Boundary-layer theory, 7th edition. New York, N.Y., McGraw-Hill Book Co., 1979.

Appendix

MATERIAL

Table 5. Composition of Materials

Trade Name	Composition (%)	Manufacturer
INCO 625	61.0 N; 0.05 C, 0.25 Mm, 2.5 Fe, 0.008 S, 0.25 Si, 21.5 Cr, 0.2 Al, 0.2 Ti, 9.0 Mo, 3.65 Cb + Ta	International Nickel Co.
AMPCO 45	9.7-10.9 Al, 2.0-3.5 Fe, 4.5-5.5 Ni, 1.5 max Mn, balance Cu, 0.6 max others	AMPCO Pittsburg Corp., Milwaukee, Wis.
17-7 PH	15.5-17.5 Cr, 3.0-5.0 Ni, 3.0-5.0 Cu, 0.2-0.5 Cb + Ta, 0.07 max C, 1.00 max Mn, 0.04 max P, 0.03 max S, 1.00 max Si, balance Fe	
316 SS	16.0-18.0 Cr, 10.0-14.0 Ni, 2.0/3.0 Mo, 0.03 max S, 0.08 max C, 2.00 max Mn, 0.05 max P, 1.00 max Si	
303 SS	17.0-19.0 Cr, 8.0-10.0 Ni, 0.18-0.35 S, 0.12 max C, 2.00 max Mn, 0.04 max P, 1.00 max Si	
Elgiloy	40 Co, 20 Cr, 15 Ni, 7 Mo, 2 Mn, 15 Fe, 0.15 C, 0.05 Be	Elgiloy Co., Elgin, Ill.
Alumina SC-98D	98 Al_2O_3, balance proprietary wear materials	Centerflex Technologies, Hawthorne, N.Y.

Continued

Table 5. Continued

Trade Name	Composition ([illegible])	Manufacturer
Torlon 4301	12 Graphite powder, 3 PTFE, 85 resin	AMOCO Chemicals Corp., Chicago, Ill.
Torlon 4275	20 Graphite powder, 3 PTFE, 77 resin	AMOCO Chemicals Corp., Chicago, Ill.
Torlon 7130	30 Graphite fiber, 1 PTFE, 69 resin	AMOCO Chemicals Corp., Chicago, Ill.
Torlon 4203	3 T_iO_2, 97 resin	AMOCO Chemicals Corp., Chicago, Ill.
Torlon 4347	12 Graphite powder, 8 PTFE, 80 resin	AMOCO Chemicals Corp., Chicago, Ill.

Table 6. Properties of Metals and Ceramics

Material	Density ($lb/in.^3$)	Tensile Strength (kpsi)	Compressive Strength (kpsi)	Mode of Elasticity $\times 10^6$ (kpsi)	Thermal Conductivity ($Btu/ft^2/hr/^\circ F/ft$)	Coeff. of Thermal Exp. $\times 10^6$ ($in./in./^\circ F$)	Hardness	Yield Strength (kpsi)
INCO 625 as rolled	0.305	105-160	--	30	5.7	7.1	$26R_C$	60-110
AMPCO 45	0.272	118	--	17	0.09	9	$98R_B$	75
17-7 PH	0.276	175-200	150	29	9.7	5.6	--	155-175
303 SS	0.29	115 (annealed)	--	28	9.7	9.6	$160R_B$	35-75
316 SS	0.29	84	--	28	9.4	8.9	$79\text{-}149R_B$	30-40
Elgiloy	0.30	110 $\sim$ 360	--	30	7.2	8.8	--	70 290
Alumina SC-98D	3.86	34	430	51.6	0.071	6.9	79RN	--
Metcar	2.45 g/CL	9	43	4.2	--	--	90 shore	--

Table 7. Properties of Plastics

Material	Density ($lb/in.^3$)	Tensile Strength (kpsi)	Compressive Strength (kpsi)	Hardness	Thermal Conductivity ($Btu/hr/ft^2/^{\circ}F/in.$)	Water Absorption	Shear Strength (kpsi)
Torlon 4203	0.05	27	40	E78 (R_E)	1.7	0.28	18.5
Torlon 4301	0.05	20	30	M109 (R_M)	2.5	0.22	16.1
Torlon 4275	0.05	18.1	20	--	--	0.19	--
Torlon 4347	0.05	19.7	--	--	--	--	--
Torlon 7130	0.05	29.8	--	--	--	0.22	--

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NAVFORCARIB Commander (N42), Puerto Rico

NAVHOSP PWD - Engr Div, Beaufort, SC

NAVMAG PWD - Engr Div, Guam, SCE, Guam, SCE, Subic Bay, R.P.

NAVOCEANO Library Bay St. Louis, MS

NAVOCEANSYSCEN Code 09 (Talkington), San Diego, CA, Code 4473 Bayside Library, San Diego, CA; Code 4473B (Tech Lib) San Diego, CA, Code 5204 (J. Stachiw), San Diego, CA; Code 5214 (H. Wheeler), San Diego CA, Code 5221 (R Jones) San Diego Ca, Code 523 (Hurley), San Diego, CA, Code 5322 (Bachman) San Diego, CA, Code 6700, San Diego, CA, Code 811 San Diego, CA, Hawaii Lab (R Yumori) Kailua, HI, Hi Lab Tech Lib Kailua HI

NAVORDMISTESTFAC PWD - Engr Dir, White Sands, NM

NAVORDSTA PWO, Louisville KY, Security Offr, Indian Head MD, Security Offr, Indian Head MD

NAVPETOFF Code 30, Alexandria VA

NAVPETRES Director, Washington DC

NAVPGSCOL C. Morers Monterey CA, E. Thornton, Monterey CA

NAVPHIBASE CO, ACB 2 Norfolk, VA, Code S3T, Norfolk VA, Harbor Clearance Unit Two, Little Creek, VA, SCE Coronado, SD,CA

NAVRADRECFAC PWO, Kami Seya Japan

NAVREGMEDCEN Code 29, Env. Health Serv. (Al Bryson) San Diego, CA

NAVHOSP CO, Millington, TN

NAVREGMEDCEN PWD - Engr Div, Camp Lejeune, NC, PWO Portsmouth, VA, PWO, Camp Lejeune, NC

NAVREGMEDCEN PWO, Okinawa, Japan

NAVREGMEDCEN SCE, SCE San Diego, CA, SCE, Camp Pendleton CA, SCE, Guam, SCE, Newport, RI; SCE, Oakland CA

NAVREGMEDCEN SCE - Yokosuka, Japan

NAVSCOLCECOFF C35 Port Hueneme, CA, CO, Code C44A Port Hueneme, CA

NAVSCSOL PWO, Athens GA

NAVSEASYSCOM Code OOC-D, Washington, DC, Code PMS 395 A 3, Washington, DC, Code PMS 395 A2, Washington, DC, Code PMS 396.3311 (Rekas), Wash., DC, Code SEA OOC Washington, DC, PMS-395 A1, Washington, DC, PMS395-A3, Washington, DC, SEA 04E (L. Kess) Washington, DC, SEA-5433, Washington, DC

NAVSEC Code 6156D, Washington, DC, Code 6157D, Washington, DC

NAVSECGRUACT PWO, Adak AK, PWO, Edzell Scotland, PWO, Puerto Rico, PWO, Torri Sta, Okinawa, Security Offr, Winter Harbor ME

NAVSECSTA PWD - Engr Div, Wash., DC

NAVSHIPYD (Code 452) Shop 02 Portsmouth, VA, Bremerton, WA (Carr Inlet Acoustic Range), Code 134, Pearl Harbor, HI, Code 202.4, Long Beach CA, Code 202.5 (Library) Puget Sound, Bremerton WA, Code 280, Mare Is., Vallejo, CA, Code 280.28 (Goodwin), Vallejo, CA, Code 380, Portsmouth, VA, Code 382.3, Pearl Harbor, HI, Code 400, Puget Sound, Code 410, Mare Is., Vallejo CA, Code 440 Portsmouth NH, Code 440, Norfolk, Code 440, Puget Sound, Bremerton WA, Code 453 (Util. Supr), Vallejo CA, L.D. Vivian, Library, Portsmouth NH, PW Dept, Long Beach, CA, PWD (Code 420) Dir Portsmouth, VA, PWD (Code 450-HD) Portsmouth, VA, PWD (Code 453-HD) SHPO 03, Portsmouth, VA, PWD (Code 460)

Portsmouth, VA; PWO, Bremerton, WA; PWO, Mare Is.; PWO, Puget Sound; SCE, Pearl Harbor HI; Tech Library, Vallejo, CA

NAVSTA Adak, AK; CO Roosevelt Roads P.R. Puerto Rico; CO, Brooklyn NY; Code 4, 12 Marine Corps Dist, Treasure Is., San Francisco CA; Dir Engr Div, PWD, Mayport FL; Engr. Dir., Rota Spain; Long Beach, CA; Maint. Cont. Div., Guantanamo Bay Cuba; PWD (LTJG P.M. Motolenich), Puerto Rico; PWD - Engr Dept, Adak, AK; PWD - Engr Div, Midway Is.; PWO, Keflavik Iceland; PWO, Mayport FL; SCE Guam; SCE, Pearl Harbor HI; SCE, San Diego CA; SCE, Subic Bay, R.P.; Security Offr, San Francisco, CA; Utilities Engr Off. Rota Spain

NAVSUPPACT CO, Naples, Italy; PWO Naples Italy

NAVSUPPFAC PWD - Maint. Control Div, Thurmont, MD

NAVSUPPO PWO, La Maddalena, Italy

NAVSURFWPNCEN PWO, White Oak, Silver Spring, MD

NAVTECHTRACEN SCE, Pensacola FL

NAVTELECOMMCOM Code 53, Washington, DC

NAVUSEAWARENGSTA Security Spec (Code 01A) Keyport, WA

NAVWPNCEN Code 24 (Dir Safe & Sec) China Lake, CA; Code 2636 China Lake; Cmdr, China Lake, CA; Code 26605 China Lake CA; Code 623 China Lake CA; PWO (Code 266) China Lake, CA; ROICC (Code 702), China Lake CA

NAVWPNEVALFAC Technical Library, Albuquerque NM

NAVWPNSTA (Clebak) Colts Neck, NJ; Code 092, Colts Neck NJ; Code 092, Concord CA; Code 092A, Seal Beach, CA

NAVWPNSTA PW Office Yorktown, VA

NAVWPNSTA PWD - Maint. Control Div., Concord, CA; PWD - Supr Gen Engr, Seal Beach, CA; PWO Charleston, SC; PWO, Seal Beach CA; Security Offr, Colts Neck NJ; Security Offr, Concord CA

NAVWPNSUPPCEN Code 09 Crane IN

NCBU 405 OIC, San Diego, CA

NCTC Const. Elec. School, Port Hueneme, CA

NCBC Code 10 Davisville, RI; Code 15, Port Hueneme CA; Code 155, Port Hueneme CA; Code 156, Port Hueneme, CA; Code 25111 Port Hueneme, CA; Code 400, Gulfport MS; Code 430 (PW Engrng) Gulfport, MS; Code 470.2, Gulfport, MS; NEESA Code 252 (P Winters) Port Hueneme, CA; PWO (Code 80) Port Hueneme, CA; PWO, Davisville RI; PWO, Gulfport, MS

NCBU 411 OIC, Norfolk VA

NCR 20, Code R70; 20, Commander

NMCB 3, SWC D. Wellington; 74, CO; FIVE, Operations Dept; Forty, CO; THREE, Operations Off.

NOAA (Mr. Joseph Vadus) Rockville, MD; Library Rockville, MD

NORDA Code 410 Bay St. Louis, MS; Code 440 (Ocean Rsch Off) Bay St. Louis MS; Code 500, (Ocean Prog Off-Ferer) Bay St. Louis, MS

NRL Code 5800 Washington, DC; Code 5843 (F. Rosenthal) Washington, DC; Code 8441 (R.A. Skop), Washington DC

NROTC J.W. Stephenson, UC, Berkeley, CA

NSC Code 44 (Security Officer) Oakland, CA; Code 54.1 Norfolk, VA; Security Offr, Hawaii

NSD SCE, Subic Bay, R.P.

NSWSES Code 0150 Port Hueneme, CA

NTC OICC, CBU-401, Great Lakes IL

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ONR Central Regional Office, Boston, MA; Code 221, Arlington VA; Code 481, Bay St. Louis, MS; Code 485 (Silva) Arlington, VA; Code 700F Arlington VA; Security Offr, Arlington VA

PACMISRANFAC HI Area Bkg Sands, PWO Kekaha, Kauai, HI

PHIBCB 1 P&E, San Diego, CA

PMTC Code 3144, (E. Good) Point Mugu, CA; Code 3331 (S. Opatowsky) Point Mugu, CA; EOD Mobile Unit, Point Mugu, CA

PWC ACE Office Norfolk, VA; CO Norfolk, VA; CO, (Code 10), Oakland, CA; CO, Great Lakes IL; CO, Pearl Harbor HI; Code 10, Great Lakes, IL; Code 105 Oakland, CA; Code 110, Great Lakes, IL; Code 110, Oakland, CA; Code 120, Oakland CA; Code 128, Guam; Code 154 (Library), Great Lakes, IL; Code 200, Great Lakes IL; Code 200, Guam; Code 400, Great Lakes, IL; Code 400, Pearl Harbor, HI; Code 400, San Diego, CA; Code 420, Great Lakes, IL; Code 420, Oakland, CA; Code 424, Norfolk, VA; Code 500 Norfolk, VA; Code 505A Oakland, CA; Code 600, Great Lakes, IL; Code 610, San Diego Ca; Code 700, Great Lakes, IL; Code 700, Norfolk, VA; Code 700, San Diego, CA; Library, Code 120C, San Diego, CA; Library, Guam; Library, Norfolk, VA; Library, Pearl Harbor, HI; Library, Pensacola, FL; Library, Subic Bay, R.P.; Library, Yokosuka JA; Util Dept (R Pascua) Pearl Harbor, HI; Utilities Officer, Guam

SPCC PWO (Code 120) Mechanicsburg PA

SUPANX PWO, Williamsburg VA

TVA Solar Group, Arnold, Knoxville, TN
UCT ONE OIC, Norfolk, VA
UCT TWO OIC, Port Hueneme CA
US DEPT OF INTERIOR Bur of Land Mgmnt Code 583, Washington DC
US GEOLOGICAL SURVEY Off Marine Geology, Piteleki, Reston VA
USAF REGIONAL HOSPITAL Fairchild AFB, WA
USAF SCHOOL OF AEROSPACE MEDICINE Hyperbaric Medicine Div, Brooks AFB, TX
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